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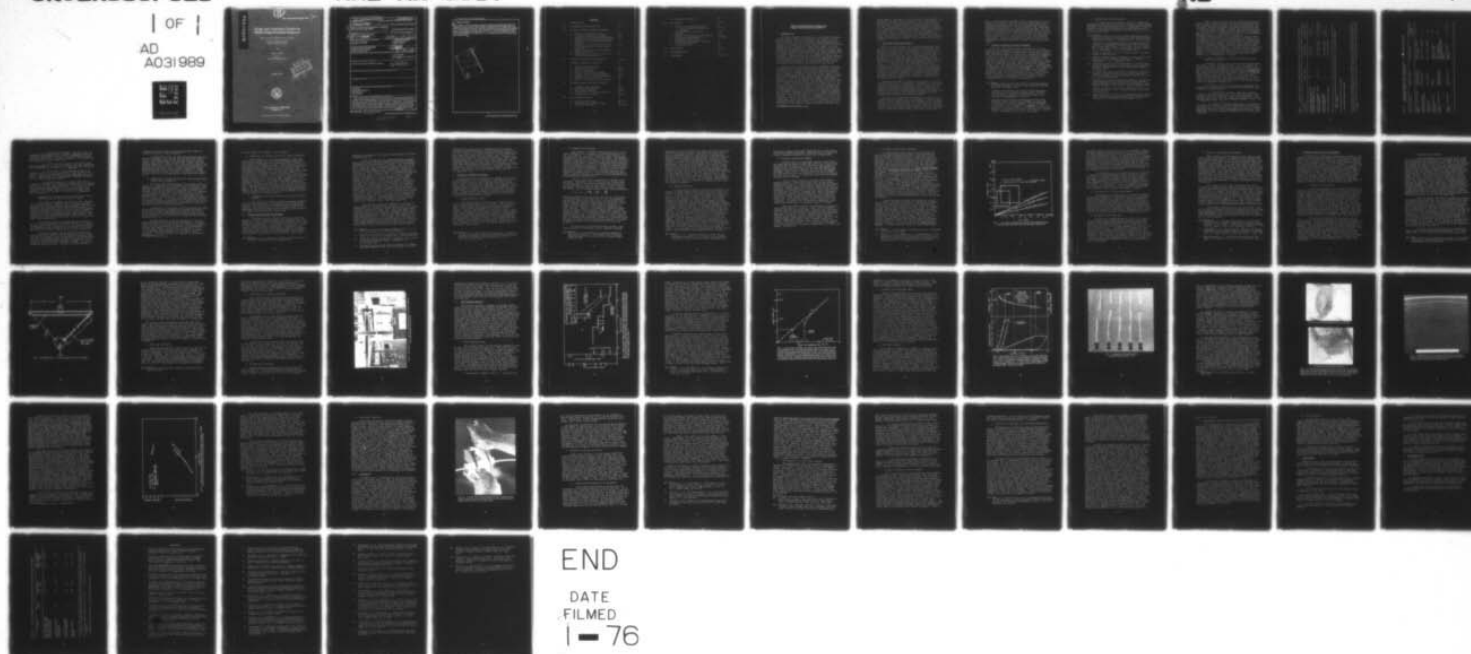OCT 76 E R SEIBERT, J M KRAFFT, H L SMITH

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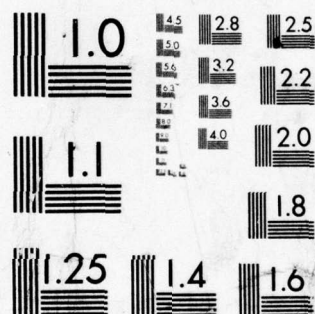| OF |
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Design and Verification Criteria for Missile Weight-Handling Equipment

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This report describes an NSPO sponsored task to evaluate the safety and structural reliability of TRIDENT-C4 missile handling equipment. In a first phase, drafts of new design specifications were examined in view of current government and industry criteria for similar service. In this second phase, design stress allowables as well as a proof loading test policy for verification of structural integrity were assessed by study and experimental programs. Structural elements of mild steel and aluminum alloy weldments and of steel wire rope are taken to typify this service. Potential sources of initial and/or use-related failure include welding defects, cold cracking, lamellar tearing, fatigue, and salt

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20. Abstract (Continued)

water corrosion. A structural specimen consisting of a spreader beam and wire rope pendants was designed, then tested in static and fatigue loading, with simulated proof test loadings; both air and salt water environments were imposed. Failure of the spreader beams was caused by crack propagation from a weld, with fatigue life consistent with literature data. The wire rope pendants also failed by fatigue crack propagation in the wires adjacent the core. Here the proof test overload was found to delay the fatigue damage, hence lengthen its life. A new technique is proposed for detecting broken wires inside the rope.

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DESIGN AND VERIFICATION CRITERIA FOR
MISSILE WEIGHT-HANDLING EQUIPMENT

I. INTRODUCTION

The Navy, through contractors to its Strategic Systems
Project Office, NSPO, is developing a new system of sub-
marine-launched strategic missiles. The prime objective of
the new system is to obtain greater maximum range/payload.
A missile configuration, larger than the current C-3, called
TRIDENT C-4, is required to obtain this range. The new,
larger missile requires new, larger handling and servicing
equipment. Preparatory to acquiring the new equipment, NSP
has contracted with Lockheed Missile and Space Company, LMSC,
to prepare specifications for the design and test of the
equipment. Concurrently, NSP 2720 tasked the Naval Research
Laboratory (NRL Code 8403) to provide an independent review
of these specifications prior to their acceptance for use
in procurement contracts. The first part of this report
summarizes the procedures and results of this review.

In review of the new specification documents, controls
are applied in the conventional manner, by restricting the
stresses due to loading, the allowable stresses, to some
level that will clearly obviate gross plastic deformation
and/or inelastic buckling. Hence the study compares allow-
able stresses among design practices for the various classes
of service involved in the missile support requirements.
However, it must be recognized that simply restricting the
loading stresses cannot assure freedom from structural fail-
ure. Cracks can initiate and grow to critical size at low
stress levels with little effect on the functioning of the
equipment, right up to the point of fracture instability.
To guard against such occurrence, two procedures are used.
The equipment and/or its components are inspected visually
and with NDT techniques such as by X- or Gamma- radiation,
dye penetrant, magnetic particle, ultrasonics, and acoustic
emission. Another procedure is to apply an overload in a
controlled test to verify the load capability of the entire
assemblage - proof test certification. Because of the pot-
entially extreme consequences of structural failures, the
Navy practice has been to require frequent recertification

Note: Manuscript submitted September 22, 1976.

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of the support equipment by proof testing. The cost of such maintenance is high, both in terms of men and equipment required to process the equipment as well as in the logistics problem of transporting it to and from the test site and time off-station. This cost would be clearly justified if it assured absolutely the structural integrity. However, it is reasonable to occasionally reassess the value of such procedure, in view of advances in our understanding of failure processes, particularly in the mechanics of fracture. This question is addressed in a further NSP-sponsored study and experimental program at NRL, which is also reported here.

II. EQUIPMENT TYPES AND SERVICE

The new missile system and its handling equipment will, of course, not be all new. As in most technological advances, it is built upon past experience. Hence, in examining specifications for the new system, as in devising a "typical" component for a test program, cognizance was taken of earlier missile systems, e.g., the Poseidon C-3 and Polaris A-3. A survey of about a hundred LMSC drawings showing various items of weight-handling equipment served to identify salient features which the equipment should contain. It is notable that the equipment encompasses a variety of configurations, sizes and functions. They are classified according to function: lifting; handling; and transporting. Lifting equipment includes hoisting machines as well as auxiliary tackle such as slings, lifting beams and bands. Handling equipment includes assembly stands, erecting fixtures, cradles and containers. Transporting equipment includes dollies and trailers designed for use afloat as well as ashore.

Although a variety of devices are involved, it is notable that each piece of equipment is designed for the handling and servicing of a particular item - none of it can be classed as of general-purpose use. This is important in assessing the degree of conservatism provided by a given factor of safety or design-allowable stress as it means that the weight of objects to be handled is known much more closely than with general purpose equipment.

The need for recertification depends on sources of service related deteriorations. Central here is the use frequency. Design specifications for TRIDENT support equipment are based on a service life of 10 to 20 years. An extreme service use would require that a unit be used for one cycle on each working day which sums to about 6000 total operating cycles. With this under present recertification policy, a

piece of shore-based equipment would be subjected to 130 use
cycles in the 6-month interval between proof loads, while
ship-based equipment, 780 cycles in the 36-month interval
between recertifications. By any standard, this represents
a moderate requirement for repeated-load fatigue. An auto-
mobile connecting rod accumulates this many cycles in a few
minutes of running time: a helicopter rotor, in less than an
hour. Nonetheless, it is still a possible source of use-
related deterioration, particularly in the salt air/water
environment. Hence the analysis of safety as well as the
experimental program emphasize fatigue in the at-sea envir-
onment.

III. REVIEW OF DESIGN CRITERIA DOCUMENTS

Three new specifications were reviewed in the first
phase of this assignment: "General Specification for Missile
Body Handling and Servicing Equipment, Trident Missile Sys-
tem," NAVORD WS 13963, 31 August 1973, designated as Spec 63
in the text [1]*; "General Specification for Missile Body
Containers, (Reusable) and Transportation Equipment, Tri-
dent Missile System," NAVORD WS 13964, 31 August 1973, des-
ignated as Spec 64 [2]; and "General Specification for Hand-
ling and Servicing Equipment, MK4 Re-entry Body Assembly,
Trident and Fleet Ballistic Missile Systems," NAVORD WS 13969
21 September 1973, designated as Spec 69 [3]. The immediate
recommendations resulting from this review were promptly
reported to NSP Code 2720 (December 1973) in the form of
annotated copies of the specifications and a letter report
dated 28 May 1974 containing tentative design criteria.
A more permanent documentation of the basis for those
recommendations is provided in this section.

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1. "General Specification for Missile Body Handling and
    Servicing Equipment, Trident Missile System," NAVORD
    WS 13963, 31 August 1973.

2. "General Specification for Missile Body Containers,
    (Reusable) and Transportation Equipment, Trident
    Missile System," NAVORD WS 13964, 31 August 1973,
    (designated as Spec 64 in the text).

3. "General Specification for Handling and Servicing
    Equipment, MK4 Re-entry Body Assembly, Trident and
    Fleet Ballistic Missile Systems," NAVORD WS 13969,
    21 September 1973 (designated as Spec 69 in the text).

*   Numerals in brackets refer to references listed at the
    bottom of each page where first cited and in a complete
    list of references at the end of the report.

1. Relevant Documents Reviewed

   To prepare for this study, a number of definitive documents on handling equipment were studied. These include standards and instructions of the U.S. Department of Defense and Labor, and technical manuals and specifications published by industrial associations and institutes [4, 5, 6, 7, 8, 9, 10, 11, 12].

4. "General Design Criteria for Handling Equipment Associated with Weapons and Weapons Systems," MIL-STD-1365, Department of Defense, Washington, D.C. 27 April 1972.

5. "Policies and Procedures for the Utilization and Development of Standards Relating to Accident Prevention and Occupational Health," OPNAV Inst. 5100.17, Department of the Navy, Office of the Chief of Naval Operations, Washington, D. C., 6 September 1972.

6. "General Specification for Aircraft Slings," MIL-STD-5944B, 12 January 1973.

7. "Metallic Materials and Elements for Aerospace Vehicle Structures," MIL-HDBK-5B, Department of Defense, Washington, D. C., 1 September 1971.

8. "Inspection and Proof Load Testing of Lifting Slings and Restraining Devices for Aircraft and Related Components," NAVAIR-17-114, Naval Air Systems Command, 15 March 1972.

9. "Occupational Safety and Health Standards: Industrial Slings," Occupational Safety and Health Administration, Department of Labor, Federal Register, Vol. 38, No. 168, 30 August 1973.

10. "Occupational Safety and Health Standards: Safety and Health Regulations for Construction," Occupational Safety and Health Administration, Department of Labor, Federal Register, Wol. 37, No. 243, 16 December 1972.

11. "Manual of Steel Construction," American Institute of Steel Construction, Inc., Seventh Edition, New York, New York, 1973.

12. "Specifications for Electric Overhead Traveling Cranes," Crane Manufacturers Association of America Specification No. 70, Pittsburgh, Pennsylvania, 1971.

Some of these sources must be ranked above others in view of their adoption by DOD. Firstly, MIL-STD-1365 [4], which contains general design criteria and applicable tests for weight handling equipment associated with nuclear and non-nuclear weapons systems, contains the following notice:
"This military standard has been approved by the Department of defense and is mandatory for use by all departments and agencies of the Department of Defense".
Secondly, OPNAVINST 5100.17 [5] requires the Navy to meet or exceed the provisions of the published standards of the Occupational Safety and Health Administration. Exceptions are permitted only where a unique military situation exists or where available evidence indicates that existing facilities or equipment provides protection equivalent to that required by the OSHA standards. Thirdly, the aircraft sling specification MIL-STD-5944B [6] covers the general requirements, including structural design criteria, for aircraft slings. It is approved for use by all departments and agencies of the Department of Defense.

2. A General Comparison of Design Criteria for the New Specifications

A detailed study of the three weapons specifications has shown a number of similarities. As is reasonable, a greater degree of consistency is observed between the missile and re-entry body handling specifications, Specs 63 and 69; a lesser degree with that of their containers, Spec 64. The structural design loads and safety factors cited in all specifications are shown in Table I. These, and other common requirements, are recited below as points of information for the present. They are considered here in conjunction with corresponding criteria of other design authorities, as summarized in Table II.

A variety of service environmental conditions are designated. Observations on particular environmental specifications are noted below.

a) Natural environments: identical climactic conditions are specified in all three specifications: temperature, pressure, humidity, rain, salt atmosphere, wind, sand and dust, sleet and hail (Specs 63 and 69, Tables I and II, Spec 64, Tables V and VI).

b) Shipboard dynamic conditions: A capability to withstand and function under identical severities of shock and vibration, roll and pitch is called out for both missile (Spec 63) and re-entry body (Spec 69) specifications. However, the container equipment is governed by differing requirements.

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TABLE I. LOADS AND FACTORS IN SPECIFICATIONS WS 13963, WS 13964, and WS 13969

Equipment	Limit Load	Yield Load	Ultimate Load
ALL EQUIPMENT OTHER THAN HOISTING EQUIPMENT & UNFIRED PRESSURE VESSELS	*L	1.15 X L 1.5 X L	1.5 X L 2.0 X L (1)
HOISTING EQUIPMENT EXCEPT CABLE ASSEMBLIES	2.0 X W	1.15 (2.0 X W) 1.5 (2.0 X W)	1.5 (2.0 X W) 2.0 (2.0 X W) (1)
HOISTING EQUIPMENT CABLE ASSEMBLIES	W	NOT APPLICABLE	5 X W
UNFIRED PRESSURE VESSELS	P	NOT APPLICABLE	6 X W (1)

L = LIMIT LOAD OR MAXIMUM LOAD EXPECTED DURING NORMAL LIFE INCLUDING DYNAMIC LOAD FACTORS

M = MAXIMUM WEIGHT TO BE HOISTED

P = MAXIMUM WORKING PRESSURE

*For dollies and equipment moved or transported in the loaded condition by crane, truck, forklift, or straddle carrier the minimum vertical limit load shall be 1.5W.

(1) Loads and factors presented in revision of WS13969, WS13969A dated 19 September 1975.

TABLE II. VARIOUS MINIMUM SAFETY FACTORS BASED ON YIELD STRESS, HOISTING EQUIPMENT, EXCEPT CABLE ASSEMBLIES.

Type of Service	Source - Ref.	Design Load	"Factor" of Safety
Missile and Re-entry Body	Proposed Navy Specifications	Max. Static Weight + Friction	2.3
Weapons Systems (Includes Nuclear)	MIL-STD-1365	Max. Static Weight	5.0
Aircraft and Components	MIL-S-5944B	Max. Static Weight	3.0
Weapons Handling-Shipboard	NAVSEC	Static Weight + Dynamic Load	2.9
General Industrial Service	Manufacturing Firm (A.I.S.C.)	Static Weight + 25% Dynamic Weight + 25% Allowance for Impact	≈ 2.1
Crane - Structural Design	C.M.A.A. Spec. No. 70	Static Weight + Dynamic	≈ 1.7 to 2.0

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c) Transportation dynamic conditions: For the container and transportation equipment (Spec 64) shock and vibration requirements differ for various modes of transportation: road, rail, aircraft, shipboard. Containers are also required to pass a drop test.

d) Limit load: It is used as the design load for all equipment. It includes dynamic loads for containers and transportation equipment (e.g., Spec 63, Par. 3.4.11.2).

e) Yield load: The minimum yield factor of safety is set at 1.15, a value which should obviate occurrence of any plastic deformation (e.g., Spec 63, Par. 3.4. 11.3). Yield loads are not designated for cable assemblies nor for unfired pressure vessels.

f) Ultimate load: It is specified for all classes of equipment. The minimum ultimate factor of safety is set at 1.5. For cable assemblies it is set at 5.0 while for unfired pressure vessels, at 4.0. This load may not induce stresses larger than the ultimate strength of the material, nor cause rupture or collapse (e.g. Spec. 63, Par. 3.4.11.4).

3. Comparison of Design Criteria for Equipment Other Than Hoisting and Unfired Pressure Vessels

This class of equipment includes dollies, assembly and erection fixtures, cradles, etc. The three proposed specifications require that the limit (design) load for this equipment be the maximum load expected during normal life including dynamic load factors. For dollies and equipment moved in the loaded condition by crane, truck, forklift or straddle carrier, the minimum vertical limit load is indicated as 1.5 times the maximum weight being handled. The minimum yield and ultimate loads are 1.15 and 1.5 times the limit load respectively.

The primary source of detailed design information for this type of equipment is MIL-STD-1365 [4]. It sets a minimum yield factor of safety of 3.0 for metallic elements of handling devices, except those in lifting or hoisting. Environmental criteria dynamic tests are also specified.

The OSHA standard "Safety and Health Regulations for Construction" presents safety requirements for a wide variety of equipment used in construction, including material handling equipment such as forklift trucks and straddle carriers. The requirements pertain to inspection, test, operation and maintenance standards; however, it does not establish design criteria. However, the OSHA standard on

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"Industrial Slings" does specify maximum safe loads for different sizes of flexible members.

A comparison of the new LMSC specifications and MIL-STD-1365 indicates that the two specify differing design loads. The design loads of the new specification reflect dynamic load factors while the military standard uses the maximum weight to be lifted. When dollies and equipment are moved in the loaded condition, the new specifications set a minimum vertical limit load 1.5 times the maximum weight. Multiplying this by the yield factor of safety 1.15 indicates an effective factor of safety on yield of 1.72 which is less than the 3.0 recommendation of MIL-STD-1365 (Table II).

4. Comparison of Design Criteria for Hoisting Equipment, Except Cable Assemblies

The three LMSC specifications set minimum factors of safety of 2.3 based on yield strength, and 3.0 based on the ultimate strength of the structure. More conservative, MIL-STD-1365 requires that metallic structural components be designed with a yield factor of safety of 5.0. The standard for the design of aircraft slings, MIL-STD-5944B, sets a minimum factor of safety of 3.0. For attachment devices such as bolts, rivets and pins, an ultimate factor of safety of 5.0 is required. It is noted that the materials section of that specification, paragraph 3.4.2, requires that steels meet ASTM or AISI Standards and be inherently corrosion resistant or treated to resist corrosion.

Information on the design criteria required for shipboard weapons handling equipment was provided by James M. Coffin* of the Naval Ship Engineering Center. Here structural components are designed with a yield (stress) factor of safety of 2.9. The same factor is required for buckling failure (Table II).

Inquiry was made of design engineers in some key industrial activities involved in crane design. A considerable reliance on Crane Manufacturer's Association Specification No. 70 is indicated, in addition to the generally utilized AISC Manual. Yield factor of safety of 1.7 to 2.0 are common, with allowance for loading due to impact (15 to 50%) and accelerational forces, some of them torsional. Allowable stresses are reduced if the service

* Private communication.

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involves repetitive loading, hence fatigue.

5. Comparison of Design Criteria for Cable Assemblies

There appears to be general agreement among the authorities checked that cable assemblies employing wire rope should be designed with a minimum factor of safety of 5.0 relative to breaking strength. Other materials such as alloy steel chain, synthetic fiber rope and steel mesh are commonly afforded even larger factors of safety. The requirements of MIL-STD-1365 [4], are more conservative than the others with respect to steel wire rope (Table II); a minimum factor of safety of 6.25 is indicated. The aircraft sling specification (Sec. 2.a.) requires a minimum factor of safety of 5.0 as do the proposed OSHA standards [9], the rigging handbook [13], the Crane Specification [12] and the engineers contacted in this study. The OSHA standard on industrial slings as approved and issued deletes requirement that a minimum factor of safety of five be used for wire rope. The OSHA standard on Industrial Slings contains **information** on the permissible load for slings manufactured of various materials as a function of size of member, orientation and number of lifting elements.

6. Comparison of Design Criteria for Unfired Pressure Vessels

This type of equipment includes basically shipping containers which provide a low-pressure controlled-atmosphere for the protection of equipment during shipping and storage. The design limit of P and ultimate loads of 4.0 P prescribed by the three LMSC specifications are in keeping with ASME (unfired) pressure vessel code.

IV. CURRENT VERIFICATION REQUIREMENTS

Proof-load testing often serves as the least expensive, and most reassuring method of verifying structural integrity. There are many engineering structures where this method is not possible. One can hardly proof-load a bridge, a steel building frame, a large ship. In aircraft, so valued is the proof test that the most elaborate, costly, full scale load tests are undertaken as part of the verification procedure. Thus it is understandable with weight-handling gear, where the provision of extra load for proof

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13. Rossnagel, W. E. "Handbook of Rigging," Third Edition, New York, McGraw-Hill Book Company, 1964.

testing is so natural, that proof load testing is a widely
accepted practice.

At the present time all weight-handling equipment
used in servicing submarine-launched missiles and missile
components must be proof-tested as part of the acceptance
and periodic recertification procedures. The size of the
proof loads and their frequency of application are speci-
fied in Ordnance Pamphlet OP-4 "Ammunition   Handling
Afloat [14], and Ordnance Pamphlet OP-5 "Ammunition Hand-
ling Ashore" [15]. General ordnance safety precautions are
contained in NAVSEA OP 3347 [16]. For types of weight-
handling equipment of primary interest in this study -
slings, strong backs, spreader beams which interface
between the item to be lifted and the cranes, and hoists -
OP-4 and OP-5 require proof testing every six months for
equipment used ashore and every 18 months for equipment
used afloat. A discrepancy exists in that the test fre-
quency for gear used afloat is indicated as only every
three years in OP-4,every 18 months in NAVSEA OP 3347 [16].
The level of proof loading is 2.0 to 2.15X of their static
rated load. This load is to be held for two minutes for
metallic materials and for five minutes for non-metallic
load-bearing members. The magnitudes of proof loads re-
quired for cranes and hoists expressed in terms of the
rated static load are lower - all less than 2.0X.

The Navy proof test requirements are closely rel-
ated to those of the Department of Labor. Proof test
requirements for industrial slings are treated in the OSHA
Standard "Industrial Slings" [9]. Executive Order 11612,
of 26 July 1971 [17], states that safety and health pro-
grams established in the Federal Government shall comply
with OSHA requirements. Navy policy, with reference to
OSHA requirements [5], is --"to apply, and develop as nec-
essary, safety and health standards equal to, or better
than, those promulgated for the private sector". The pro-
posed OSHA standard on industrial slings [9] called for
proof loads of wire rope and alloy steel chain slings of

14.   "Ammunition Afloat," Ordnance Pamphlet OP-4, Vol. 2,
       Rev. 5, February 1972, 0630-LT-000-1220.

15.   "Ammunition and Explosives Ashore," Ordnance Pamphlet
       OP-5, Vol. 1, Rev. 4, December 1974, 0630-LT-000-2121.

16.   "Ordnance Safety Precautions," NAVSEA OP 3347, United
       States Navy Technical Manual, Washington, D. C.
       15 February 1972.

17.   "Occupational Safety and Health Programs for Federal
       Employees, Executive Order 11612, White House, 26 July 1971.

200% which was essentially in agreement with the other requirements affecting proof loading magnitude, Spec (8, 14, 15). However, the approved version of the Standard has been published in the Federal Register [18] and omits any proof test requirement for wire rope slings. However, an initial proof test is required for alloy steel chain slings ASTM specification A 391-65, which prescribes proof and breaking loads based on chain size, is called out. It is noted that OSHA standard does allow for circumstances under which proof loading can be omitted. Generally a long record of no proof test failures may be taken to assure the initial quality of a wire rope product (e.g., Sec. 6, p. 27369, ref. [9]).

V. POTENTIAL SOURCES OF FAILURE

In order to assess the adequacy of design and verification requirements, it is necessary to know what sort of defects could exist in new equipment; what sorts could result from its service use. The matrix of possible defects vs types of structural components is extensive. It is attempted here to select for detailed assessment the more common types of elements which are critical in the load path. In particular, the elements selected are weldments of mild steel and aluminum alloy/structural sections or plate and wire rope tension members.

1. Common Welding Defects

The initial quality of the weld is basic to the service life expectancy of the structure. Poor welds can lead to early crack initiation, even contain crack-like flaws. Weld defects are generally classifed in five groups: 1) lack of penetration or fusion; 2) voids or porosity; 3) non-metallic inclusions; 4) cracks; and 5) imperfect shape leading to stress concentration. In addition, thermal contraction during weld solidification generally leaves residual (tensile) stresses in and around the weld. Non-destructive testing of critical welds should lead to the location and rejection of poor welds. Indeed it is the proof loading test, as one of the NDT procedures, which is assessed in this project.

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18. "Occupational Safety and Health Standards: Industrial Slings" Occupational Safety and Health Administration, Department of Labor, Federal Register, Vol. 40, No. 125, 27 June 1975.

2. Delayed Cold Cracking

One concern is delayed cracking which could occur after welding so as to weaken the structure before it is put into service. In steel weldments, three factors are recognized as contributory to cold cracking. The hydrogen content of the weld is a factor. This can be controlled by using low-hydrogen dry electrodes, gas shielding, and pre-heating of the work. Secondly, susceptible micro-structure such as untempered martensite is avoided by limiting heat input and interpass temperature. Thirdly, residual shrink-age stresses drive the delayed cold crack, and the joint constraint, volume of weld metal, welding sequence, and degree of preheat affect it. Generally welding engineers and/or qualified welders are aware of this problem and use weld procedures which avoid it.

Not all steels are susceptible to delayed cold cracking. Generally, factors which lead to higher resid-ual stress such as high yield strength and/or propensity to form martensite, such as high carbon content, are un-favorable. Various compositional criteria are available for steels, one of the more useful is the carbon equivalent

$$C. E. = \%\ C + \frac{\%Mn}{4} + \frac{\%Cr}{10} + \frac{\%Ni}{20}$$

Generally steels of CE less than 0.50 are not susceptible to cold cracking, those from 0.50 to 0.57 are moderately susceptible, while those above are very susceptible and require special measures in welding. The A-36 hot rolled steel, commonly used in surface support equipment, is below CE = 0.50, hence it should not be susceptible to cold cracking. However, in weldments of higher strength steels in weight critical parts, appropriate precautions should be taken. It is generally found that whatever cold cracking occurs will be completed within 48 hours from the time of welding [19]. For this reason, it would seem pru-dent to schedule NDT acceptance procedures, including proof testing, no sooner than 48 hours after welding to be assur-ed that any possible cold cracking that can occur has occurred.

In alloys of salt-water resistant weldable alum-inum, different circumstances lead to delayed cracking.

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19. Mishler, H. W. "Prevention of Delayed Cracking in Ship Steel Weldments," Forthcoming Ship Structure Committee Report on Project SR-210, Contract NSEA 9029-003-2010.

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Generally, the heat treatment of heavy sections leaves significant residual stresses. Typically the center cools last in quenching from the solid solution temperature, leaving surface regions in compression, center in tension. Aging temperatures are not high enough to relieve these stresses. If subsequent machining cuts through the compressive surface, the tensioned material is exposed to the atmosphere. Stress corrosion cracks tend to grow along precipitate-rich bands in aluminum, which tend to stream parallel the flow directions of rolling or forging. The insidious thing here is that, unlike tempered martensite, stress corrosion cracks grow very very slowly, yet steadily, in some aluminum alloys. Thus it would be hard to prescribe an NDT waiting period which would be long enough to toally preclude post-weld cold-cracking. Fortunately, the problem is not a serious one in thinner rolled plates and structural shapes of 6061-T6 so commonly used in surface support equipment.

3. Lamellar Tearing

In structures joined by riveting or bolting, it is unusual to apply high stresses across the flow lines of a rolled or forged product, the short-transverse, weak direction. However, with joining by welding, structural elements such as brackets, frames, and struts are readily joined to the surface of a plate or shell. Here weld cooling and shrinkage can induce high warpage stresses, tending to delaminate the base plate. Subsequent loading of the structure can augment the cracking tendency, causing a bulging or even a tear-out. A typical problem area is the frame to hull-plate junction in ship-hull structures; another is the strut-to-leg juncture of offshore platform jackets. To cope with this difficulty, steel producers have learned how to grade steels relative to resistance to lamellar tearing [20]. A key factor is a low level of inclusion forming elements, particularly of sulfur and oxygen. There is no standard test for susceptibility to lamellar tearing. However, the short transverse tensile ductility is a useful indicator, reduction in area greater than 20% being generally satisfactory. It is possible to weld-shut incipient delaminations around the edges of plates, so as to permit butt welding without opening up the plate. This expedient of ship yard practice in time of

20. Porter, L. F., "Lamellar Tearing in Plate Steels, (A Literature Survey)", U. S. Steel Corp. Research Laboratory, Report 10-F-002 (018-5) 29 August 1975.

short steel supply would seem inappropriate in fabrication of missile handling equipment, since the cost of steel products of adequate quality is relatively insignificant.

4. Kinds of Use-Related Damage

We know that if an equipment passes a proof test, and other NDE procedures, then any doubt that it would pass another after a normal service duty would imply that its condition had deteriorated in service or that damage had resulted from the proof loading. For NSP surface support equipment, what sorts of deterioration can be envisaged? As noted earlier, here one might think in terms of hook up and lift once a day or so; possibly a salt air environment; possibly a cold climate.

As an initial division, consider bulk-volume vs surface effects. Structural alloys such as mild steel, improved plow steel (for wire rope) and aluminum alloys are relatively stable, in a metallurgical sense, at room temperature. Strain aging can occur in carbon steels, and age hardening in aluminum alloys. However, these effects are not important in elastic loadings, nor should they appreciably affect fatigue crack propagation. Solar radiation can affect organic materials: nylon webbing, resins for composites. High energy nuclear radiation can embrittle mild steel, but this environment is not expected in this equipment. In general for the metallic components, bulk-volume deterioration is an unlikely source of trouble.

The more likely sources of degration are surface or near-surface effects: corrosion, rust, contact abrasion, rubbing, galling, fretting, wear. There is also the microcosm of some of these processes associated with crack propagation. We need to be less concerned with detecting and assessing the grosser forms of surface deterioration, usually visual inspection will reveal these, than with the hidden damage which remote, hard to detect cracks can entail. Important questions here include: how can service use make cracks grow; what is the worst-case life expectancy if they grow; how can proof loading serve to detect the growth, to alter it?

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5. Fatigue of Mild Steel Weldments

A comprehensive investigation of the fatigue of structural steel weldments has been carried out by the British Welding Research Association, published over the years by Gurney and Maddox [21,22]. They have recently completed a collection and re-analysis of the fatigue data published in the literature between 1950 and 1972, their own data constituting the larger part of that available. The study was used to check and/or revise the British design code for welded joints BS 153 Steel Girder Bridges, and BS 2573 Permissible Stresses in Cranes. Results for a wide range of strength levels, 33 to 120 ksi (227 to 828 MNm^{-2}), showed that the high cycle fatigue life of welded joints is insensitive to the strength level of the material. As a result, the fatigue design allowables are set down as independent of yield strength, quite in contrast to static strength allowables which are proportional to yield strength. For design allowable S-N curves, regression lines are put through plots of log stress (S) vs log endurance (N). To compensate for the residual stresses in non-stress-relieved welds the allowable stress level is rotated downward by 0.815 at the 2×10^6 cycle ordinate around a 10^5 cycle point as center of rotation. The resulting curve is then translated downward in stress by two standard deviations of the composite data to set the design allowables.

Gurney and Mattox define six categories of weld joint types, B through G in order of decreasing allowable stress. The 1973 AISC Handbook [11] indicates fatigue allowables with the various weld forms. Both are compared to some current static design criteria in Fig. 1. Since fatigue life weldments tend to be independent of yield strength, the fatigue allowables plot as horizontal lines. Contrary-wise, the static stress allowables appear as sloped lines radial from the origin. For steels of less than 50 ksi (345 MNm^{-2}) yield strength, the 10^5 cycle fatigue allowables exceed the static strength allowables for all classes of weld, even for the least conservative design criterion, i.e., that proposed for SPECS 63, 64, and 69. As higher strength alloys are used, the fatigue

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21. Gurney, T. R.,"Fatigue of Welded Structures," Cambridge University Press, 1968.

22. Gurney, T. R. and Maddox, S. J.,"Determination of Fatigue Design Stresses for Welded Structures from an Analysis of Data," Metal Construction and British Welding Journal, Vol.4, November 1972, pp. 418-422.
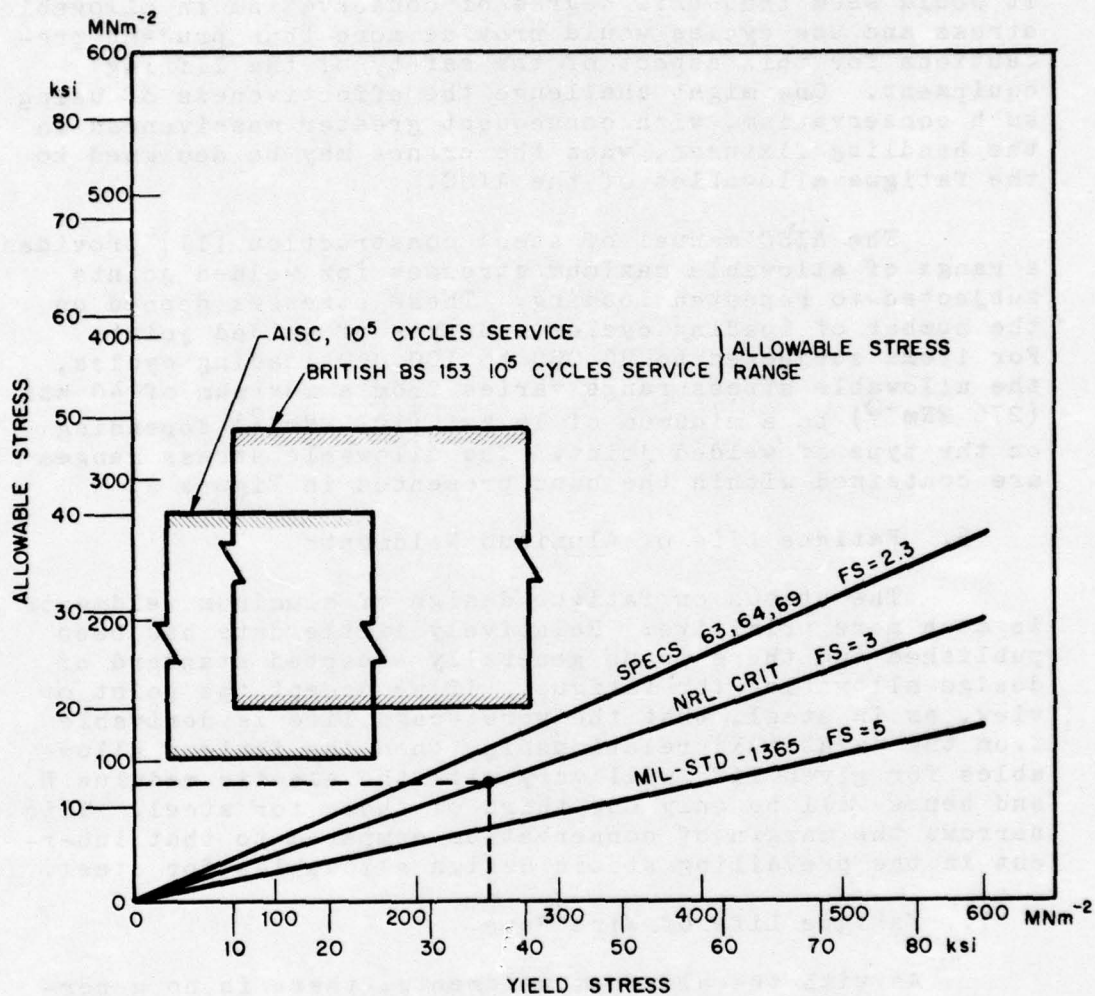
Fig. 1 — A comparison of British and American fatigue design allowables
with static design allowables indicates a conservative design policy

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criterion begins to govern the choice of allowable stress.
One might properly conclude from this that fatigue failure
is an unlikely possibility for this equipment, even with
the least conservative factor of safety for static strength.
It would seem that this degree of conservatism in allowable
stress and use cycles would provide more than prudent pre-
cautions for this aspect of the safety of the lifting
equipment. One might challenge the effectiveness of using
such conservatism, with consequent greater massiveness in
the handling fixtures, when the cranes may be designed to
the fatigue allowables of the AISC.

The AISC manual of steel construction [11] provides
a range of allowable maximum stresses for welded joints
subjected to repeated loading. These stresses depend on
the number of loading cycles and type of welded joints.
For items subjected to 20,000 to 100,000 loading cycles,
the allowable stress range varies from a maximum of 40 ksi
(276 MNm^{-2}) to a minumum of 15 ksi (103 MNm^{-2}) depending
on the type of welded joint. The allowable stress ranges
are contained within the band presented in Figure 1.

6. Fatigue Life of Aluminum Weldments

The status on fatigue design of aluminum weldments
is much more primitive. Relatively little data has been
published and there is no generally accepted standard of
design allowables for fatigue. If we accept the point of
view, as in steel, that the worst-case life is derivable
from the da/dN (ΔK) relationship, then the fatigue allow-
ables for given life will vary with the elastic modulus E,
and hence will be only one-third of those for steel. This
narrows the margin of conservatism compared to that inher-
ent in the prevailing static design allowables for steel.

7. Fatigue Life of Wire Rope

As with the aluminum weldments, there is no gener-
ally accepted design standard for fatigue of wire rope.
However, the extensive investigations of Reemsnyder can
provide a basis for estimation. It is clear from this
work that the "S-N" curve varies markedly with type of rope
and termination used. A reduction in Reemsnyder's data
points by two standard deviations for a design criteria
should not put the lower bound as a possible worst case
below the present static load design limit 1/5 the breaking
strength. This indicates no fatigue problem with 10^4 cycle
service in air. The experimental part of this program,
to be detailed later, provides a more direct assessment of
wire rope fatigue.

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8. Effect of Salt-Water Environment

This section is brief because we are aware of very little useful information. Inasmuch as both weldment and wire rope failures could be in the worst-case fatigue crack propagation, the effect of salt water on fatigue crack propagation is pertinent. It is generally believed that the ferritic lower strength steels are inherently insensitive to corrosion effects in crack propagation. Below 100 ksi yield strength, stress corrosion cracking thresholds (K_{ISCC}) tend to be quite high, i.e., near K_{Ic}. However, some recently published data of Vosikovsky [23] has shown a fairly deleterious effect of salt water on the fatigue crack propagation rate in a 65 ksi-yield strength line pipe steel. This is a research area now receiving more attention in view of offshore platform problems; it should be monitored for more developments.

With regard to wire rope components, the microstructure of improved plow steel wire, while high in carbon content (and with equivalent carbon equivalent over 0.50) is still ferritic. However the strength is high, which is generally of an unfavorable characteristic. Salt water fatigue data of Reemsnyder [24] and of Matanzo [25] shows a large degradation of fatigue life by salt water, only about 1/5 to 1/8 the endurance in ambient air. Fortunately at higher loading levels, the degradation is smaller. These effects are too large to be freely dismissed. The problem is addressed further in the experimental program, discussed later.

The weldable aluminum alloys are also rather inert on the scale of stress-corrosion cracking activity, although more information/work is needed to quantify all effects, it seems unlikely that effects of occasional exposure to shipboard environment will significantly reduce the life expectancy of these elements.

23. Vosikovsky, O., "Fatigue Crack Growth in a X-65 Line Pipe Steel at Low Cyclic Frequencies in Aqueous Environments," Trans. ASME, J. Eng. Materials and Technology, Vol. 97, Sec. H. No. 4, Oct. 1975, pp. 298-304.

24. Reemsnyder, H. S. "The Mechanical Behavior and Fatigue Resistance of Steel Wire, Strand, and Rope," Bethlehem Steel Co. Homer Research Laboratory Report, 6 June 1972.

25. Matanzo, Frank, Jr., "Axial Fatigue Testing of Wire Rope," MTS Journal, Vol 6, No. 6, Nov-Dec 1972, pp. 25-33.

VI. SPECIMEN DESIGN AND TEST PROCEDURE

An experimental program was undertaken, as part of this study, to assess the applicability of the current design and verification criteria on a "typical" assemblage of lifting gear. The item chosen is a lifting sling, designed and constructed in multiple copies as a test specimen. It consists of a spreader-beam, connected from its ends to a single load point by two convergent wire-rope pendants in isosceles-triangle array. In this way, identical loading was applied to duplicate pendants. The beam was loaded at its center by a welded tab. The entire assemblage was loaded repeatedly until failure occurred. Usually the pendants failed before the spreader beam, and were replaced as required. Both room air and salt water environments were employed. Proof testing was simulated by overloading the assemblage at regular intervals. Details follow in this section.

1. A Typical Test Specimen

How to characterize with a single "specimen" the wide variety of items actually in the C-4 system is of necessity a compromise. The selection started with a survey of a large number of engineering drawings of various items of surface support equipment of pre-C-4 systems, as noted in Sec. II. It was observed that the most common material for rigid elements was plain carbon mild steel. Hot rolled plate or structural sections of A36 is typical. In items where a lighter structure is required, aluminum alloy, particularly 6061-T6, is apt to be designated. Like the steel, 6061 is weldable without post-weld heat-treatment, although inert-gas shielding is required for its welding. The weldability of these materials is of course, consistent with the typical joining practice for permanent joints; electric arc fusion welding, using when necessary inert gas shielding.

The nature of handling/lifting operations requires, as it has from antiquity, utilization of flexible rope rigging. Fiber rope, generally of synthetic rather than natural fiber, is always available at fleet installations. However, on more specialized tools of missile handling, the flexible lines are apt to be constructed of wire rope. Here for economy and strength, improved plow steel wire is a common rope material. Alternatively, however, stainless steel rope may be employed. All things considered it was decided to design the "specimen" so as to include these basic constituents of missile handling equipment: 1) mild steel weldments; 2) 6061-aluminum weldments; and 3) steel wire rope links.

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2. Spreader Beam Assembly

The configuration selected consisted of a centrally loaded I-beam strong back, or spreader beam, with a wire rope pendant attached at each end, Fig. 2. The wire rope assemblies were inclined at 45° to the axis of the beam to increase the length of the wire rope usable within the same vertical distance. The cross-sectional properties, such as depth of beam and weight per unit length, were selected so that the aluminum and steel beams would both fail by plastic yielding before the cables would break under static tension. The assemblage was designed to provide actual factors of safety as close as practical to the use allowable, using standard size structural shapes and wire rope. The inclined reactions induce a loading parallel to the axis of the beam. This combination of loads requires that the spreader bar be considered as a beam-column for analysis purposes. However, the bending stress is critical in this configuration since the column buckling stress is relatively low and the column slenderness ratio is small. Checking the design for these combined loads followed AISC Handbook [11] for the A-36 beam and the Aluminum Association Specifications [26] for the 6061 aluminum beam.

Once the general "specimen" configuration was set, the design process started with a selection of a wire rope construction and size. This order was necessary since the allowable load on the rope is a fixed quantity while that on the beam of given section is adjustable by changing the span. Selecting a 6 x 19 improved plow steel wire rope, an allowable load of 1180 lbs, 1/5 the breaking strength was set. The vertical component of 1180 lb. at 45° load line is 834 lb. A 3-inch deep standard I-beam 5.7lb/ft, section modulus $\frac{I}{C} = 1.7$ in^3, will withstand a bending moment $SI/C = 1.7 \times 36,000 = 61,200$ in/lb at yield stress loading, or 20,400 in/lb with an effective safety factor of 3.0 on the dead weight to be lifted. 834 lb. acting on a moment arm of 24.5 inches is this moment. In actuality, a half span of 22.5 inches was chosen, increasing the effective safety factor by about 8% for the beam.

The center tab construction is the purposeful "weak link" of the spreader beam assemblage. It is fabricated from a 1/2 x 2 inch bar stock of cold rolled steel.

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26. "Specifications for Aluminum Structures," The Aluminum Association, Aluminum Construction Manual, Section 1, Second Edition, New York, 1971.
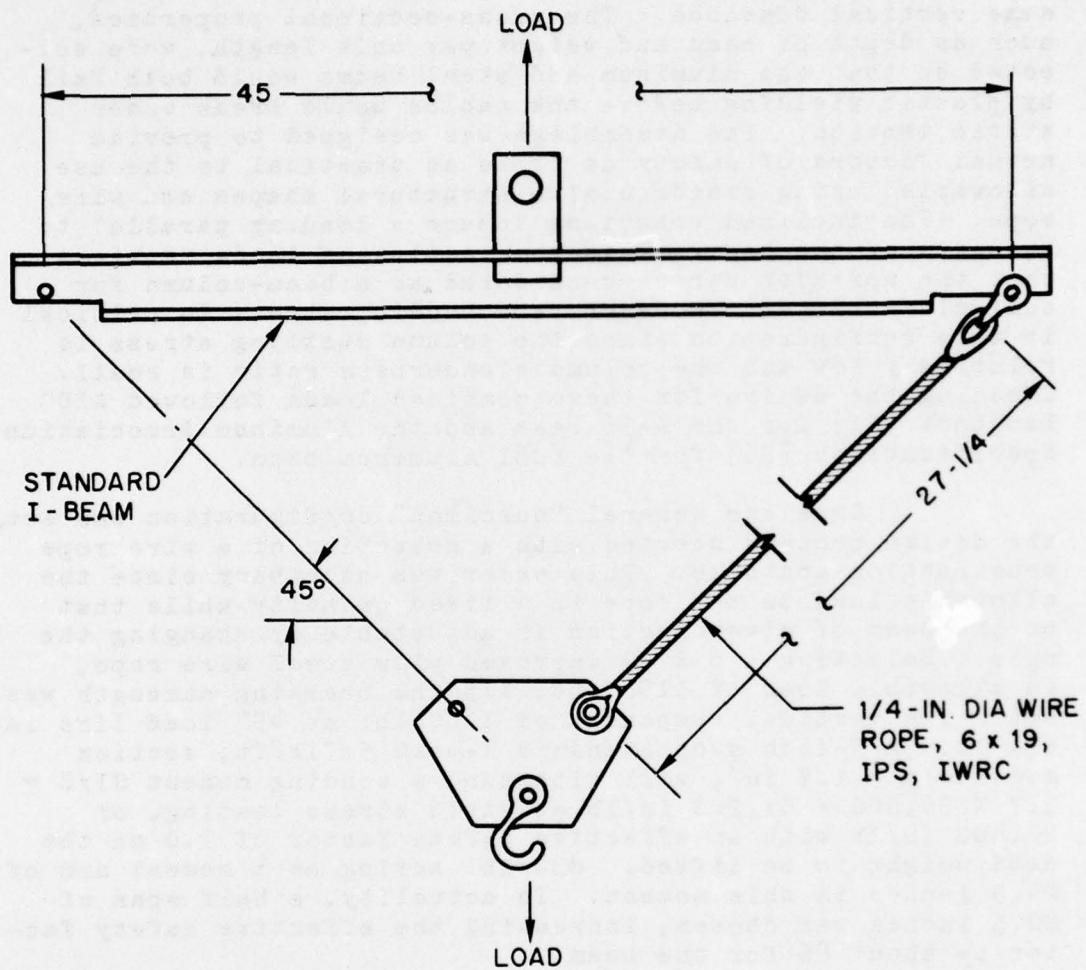
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Fig. 2 — The spreader beam — wire rope assembly used in the test program

It is slotted on one end, 1.0 inch deep, 1/4 inch wide,
which is the approximate thickness of the I-beam web. The
upper flange of the I-beam is cut away centrally with a
1/2 inch end mill, just sufficiently to permit insertion of
the slotted tongue. It is joined by 1/8 inch fillet welds
on the edges overlapping the flange as well as along its
juncture with inner and top surfaces of the flange. This
length and size of weld exceeds requirements of the AWS
Structural Welding Code [27]. Here paragraph 1.17.10
requires that fillet welds terminating at ends or sides be
turned continuously around the corner for a distance not
less than twice the nominal size of the weld. Accordingly,
the fillet welds securing the tab to the underside of the
flange and to the web were continued along the vertical
sides of the tabs and joined. Since fillet welds placed
parallel to the loading direction are not as strong as
transverse welds, the added strength due to these welds was
neglected. The allowable stress in the welds deposited,
using E60 series rods on A-36 steel, is 13.6 ksi ($94 MNm^{-2}$).
The allowable load on six two-inch long 1/8-inch fillets
was 14.4 kip (64.1KN) well over the 3338 lb. design load.

The aluminum beam was designed using the same load,
length, and angle of inclination of end loads as the steel
beam. Again a standard I-beam section was utilized, but its
cross-sectional dimensions differed because of the differ-
ence between the properties of the two metals. The proper-
ties of the section selected, a 4-inch deep standard I-beam,
3.28 pounds per foot section modulus 3.4 in^3. The larger
section depth, hence section modulus, is necessary because
solution softening due to weld heating reduces the yield
strength from the initial 35 ksi ($241 MNm^{-2}$) to about 20 ksi
($139 MNm^{-2}$).

   3. Wire Rope Assemblies

       Pendants were fabricated from 1/4 in (6.4 mm) 6 X 19
improved plow steel (IPS) wire rope of right regular lay
with an independent wire rope core (IWRC) consisting of
a total of 49 wires. Each of the strand wires is 17.0 mils
(0.432 mm) diameter; each of the 49 core wires, 11.2 mils
(0.285 mm). The pendants are terminated by thimble eyes at
each end. The eye splice involved splaying the strands of
the free end spirally back around the body of the pendant
wire, to which it was clamped by two compression sleeves

27. "Structural Welding Code," American Welding Society,
     New York, 1972.

leaving a free length of about 17.5 in (445 mm). No lubrication was added to the wire rope during the testing. Samples of the wire used in manufacturing the rope analyzed for composition: carbon - 0.48%; manganese - 0.50 to 0.60%; phosphorous - 0.014 to 0.015%; sulfur - 0.026 to 0.030%; silicon - 0.22%; balance iron.

4. Verification of Design Properties

After the principal components were designed, prototypes were tested to determine their maximum strengths and so verify the design calculations. A quantity of wire rope assemblies was then ordered from a commercial supplier. Static tests on the first of the production lot by F. R. Stonesifer showed a breaking strength of 6.79 kip (30.2 kN) vs expected minimum of 5.90 kip (26.2 kN) and rated load 1.18 kip (5.25 kN). The pendants failed in the free section, away from the eye splice.

A prototype of the steel beam was manufactured at NRL in accordance with Code 8432 Drawing No. 02-39-3. Because the beam was tested in conjunction with the cable assembly, the deflection information recorded reflected the response of all component parts, such as shackles, thimbles, rope, etc. While this precluded identifying the yield load, the ultimate load was obvious. The beam failed because of gross yielding of the outer fibers at 6.94 kip (30.9 kN) only about 8% above the calculated plastic hinge load of 6.41 kip (28.4 kN). Since the calculated load was based on the assumption that the material was perfectly elastic-plastic, neglecting the effect of strain hardening, the agreement is considered good.

A prototype of the aluminum beam was manufactured at NRL based on our Drawing No. 02-39-9. The design load is 3.34 kip (14.8 kN), the same as for the steel beam. The unit was tested by F. R. Stonesifer, and the ultimate load was found to be 7.60 kip (33.8 kN) vs a calculated yield load of 6.72 kip (29.9 kN). The failure mode involved two mechanisms simultaneously, mainly plastic hinge action with some lateral buckling of the compressive flange. The greater-than-estimated failure load indicates that the weld heat softening on the upper flange is not extensive.

5. Fatigue Test Procedure

Cyclic loading was applied in a 20 kip (89.0 kN) electro-hydraulic controlled mechanical testing machine, shown in Fig. 3. A sinusoidal load-time pattern was applied with a minimum stress near zero, i.e., $R \rightarrow 0$. Loading
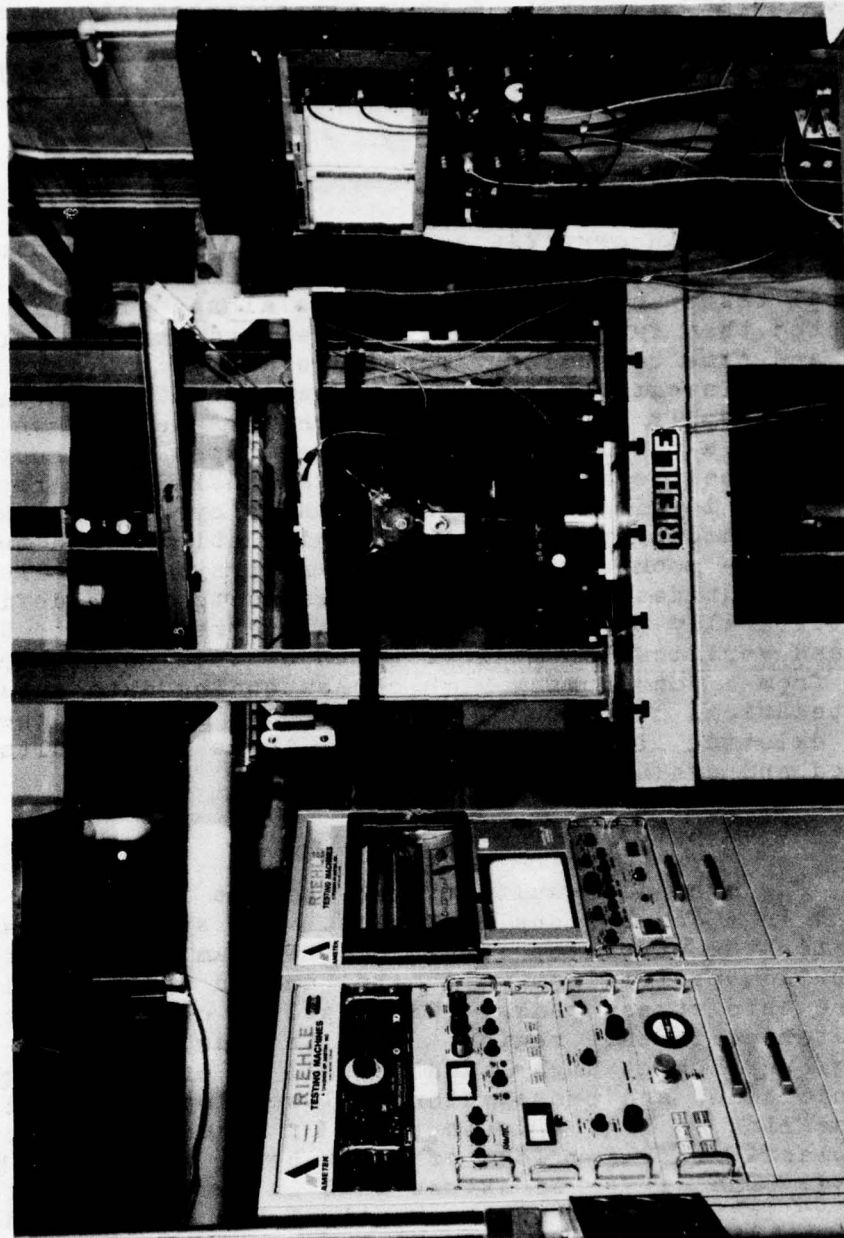
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Fig. 3 — The spreader beam — wire rope assembly being tested in an electro-hydraulic controlled mechanical testing machine

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frequency was generally limited to about 1 to 2 Hz since a substantial machine displacement was required to load the entire specimen assemblage. The overload was applied manually ten consecutive times after each block of about 4000 cycles of normal loading. A room air environment pertains to some of the data shown, the rest subjected to a salt water environment, applied by dripping or spraying synthetic sea water on the weld crack and wire pendants about every one-half hour.

VII.   TEST PROGRAM RESULTS

This section provides a summary of fatigue loading tests of the above described specimen assemblage, plus some additional tests on wire rope pendants alone. It will be appreciated that fatigue tests, as the word denotes, are tedious and time consuming. There are always too many parameters to be investigated, too few specimens and man/machine time to thoroughly explore each. Here for example, we are investigating a matrix containing three types of structural elements - steel beams, aluminum beams and IPS wire rope pendants - two loading conditions - steady cyclic load and this interrupted by proof test load simulation - and two environments - room air and salt water spray. We believe that sufficient data has been accumulated to reach definite and statistically significant conclusions relative to NSP design and verification policy. Much of this confidence derives from a consistancy with corresponding data extant in the technical literature. Accordingly, it is in the context of existing literature data that these new results are presented and evaluated.

1.   Steel Spreader Beam

The overall results of the fatigue loading is shown in Fig. 4 in a conventional S-N plot. The stress is calculated for the outer top flange of the beam, irrespective of the build up of fillet weld metal around the central tongue where this stress peaks. When failure occurred, it always started in this region. This is true of both yielding/buckling at high loads as well as of the fatigue crack initiation and propagation at lower loads. An alternative scaling in Fig. 4 is in terms of load with respect to the plastic buckling load of a new beam, about 6.90 kip (30.7 kN). The degree of excess load in the proof test simulation is indicated by the vertical bar above the data points. Recall such loads were applied in blocks of ten between each 4000 cycle block of normal loading.
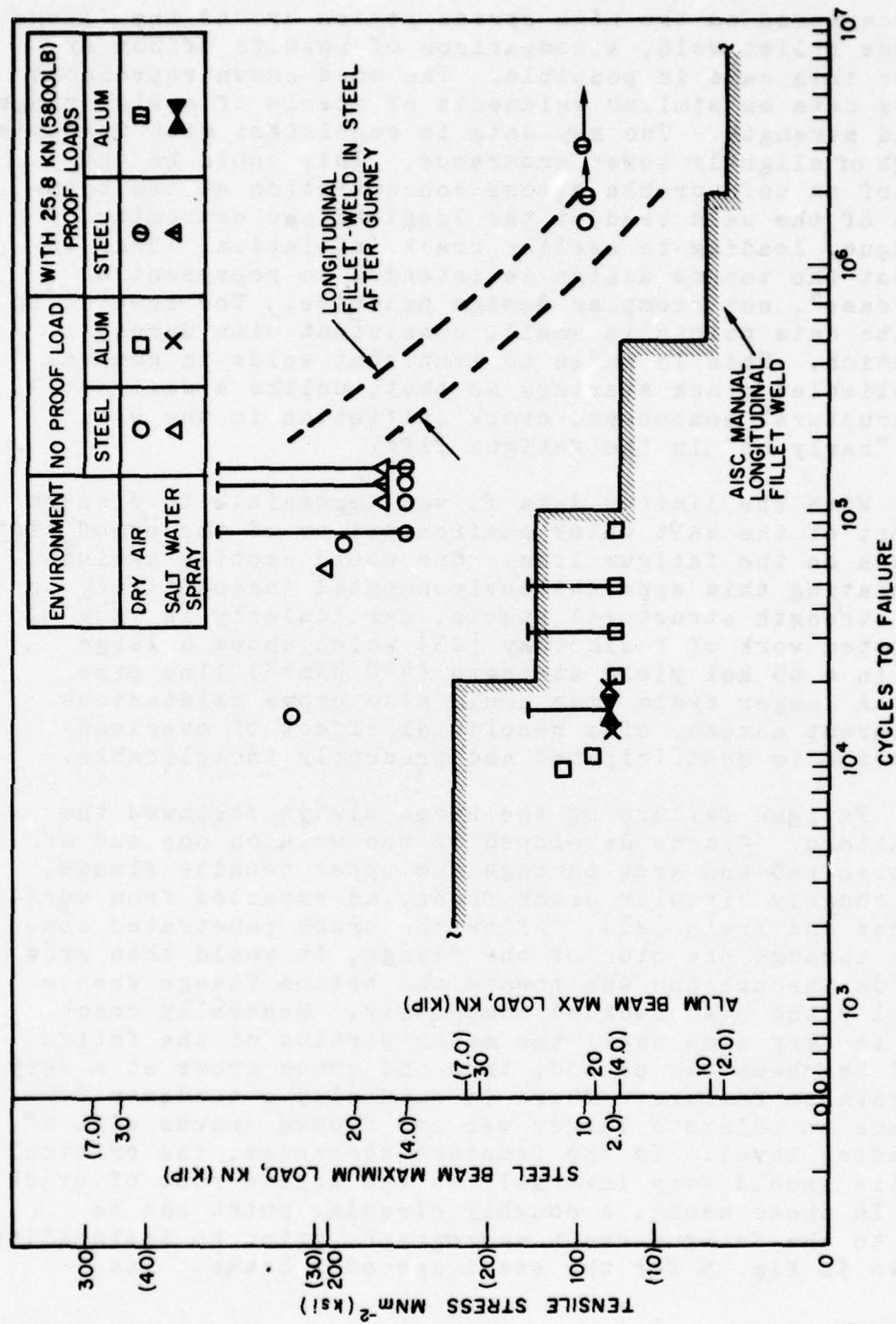
A few general comments on Fig. 5 . Since failure

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Fig. 4 — Fatigue endurance data of mild steel and aluminum spreader beams compared with test results of Gurney on welded steel members. The stresses permitted in the base metal of steel structures adjacent to fillet welds in the AISC building specifications is indicated for the range of loading cycles shown.

always centered on the high stress region around the flange to tongue fillet weld, a comparison of results of Gurney [21] for this case is possible. The band shown represents Gurney's data on similar weldments of steels of a wide range of yield strength. The new data is consistent with Gurney's, although of slightly lower endurance. This could be the result of an unfavorable stress concentration at the termination of the weld bead at the longitudinal extremity of the tongue, leading to earlier crack initiation. Bear in mind that the tongue design is intended to represent a "worst case", not exemplar design practice. The scatter in replicate data points is small, consistent with Gurney's observation. This is taken to mean that welds in general are "reliable" crack starters so that, unlike a well machined structural component, crack initiation in the weld occurs "early on" in the fatigue life.

With the limited data it was impossible to discern an effect of the salt water environment or of the proof test overloads on the fatigue life. One would caution against extrapolating this apparent environmental insensitivity to higher strength structural steels, particularly in view of above noted work of Vosikovsky [23] which shows a large effect in a 65 ksi yield strength ($448 KNm^{-2}$) line pipe steel. A longer cycle time could also prove deleterious. The apparent absence of a beneficial effect of overload retardation is unanticipated and presently inexplicable.

Fatigue failure of the beams always followed the same pattern. Cracks developed in the weld on one end of the center tab and grew through the upper tensile flange, with a roughly circular crack front, as expected from work of Fisher and Irwin [28]. After the crack penetrated completely through one side of the flange, it would then grow downwards through the web toward the bottom flange whence eventually the beam buckled completely. Generally crack growth is very slow until the major portion of the fatigue life of the beam has passed; then the crack grows at a very rapid rate to failure. There is naturally a tendency for the beams to tolerate larger web and flange cracks with a low loading level. In the fracture mechanics, the critical flaw size should vary inversely as the square root of crack size. In these beams, a roughly circular point can be fitted to the fatigue crack macrograph, prior to instability, as shown in Fig. 5 for the steel spreader beams. Its

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28. Fisher, J. W. and Irwin, G. R., "Fracture Analysis of Flaws in Welded Bridge Structures," Significance of Defects in Welded Structures, Univ. of Tokyo Press, 1974, pp. 213-223.
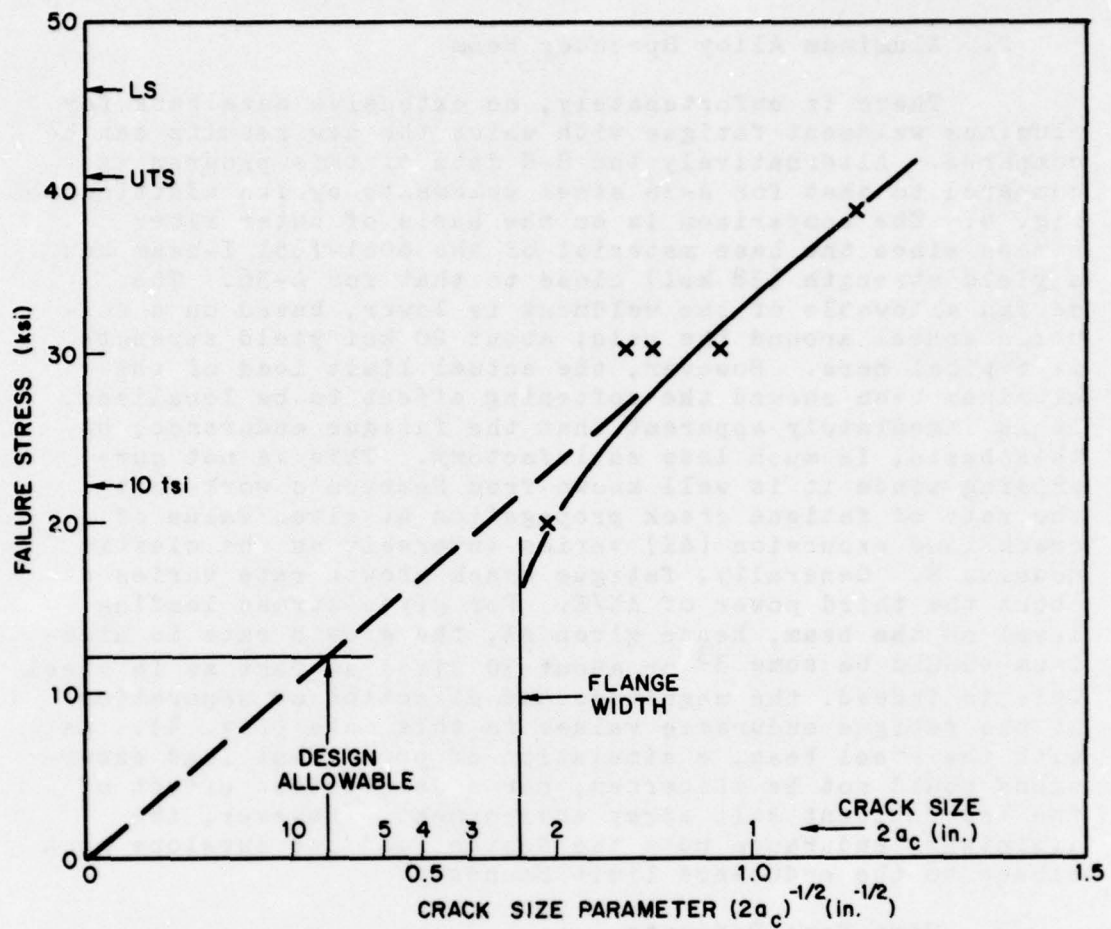
Fig. 5 — At the point of collapse, the diameter (2a) of the semi-circular crack in the flange and web of the A-36 steel spreader beam (at the collapse point) is a function of the nominal outer fiber stress in the beam. In a fracture mechanics context, proportionality to the inverse square root of crack size is expected and observed. The critical size at the design allowable (about 8 in.) is large enough to be easily detected. Alternatively rather large cracks can survive the proof test load, making it a rather insensitive NDT method for finding cracks in mild steel.

diameter to the negative 1/2 power $(2a_c)^{-\frac{1}{2}}$ in Fig. 5 does appear to correlate with the nominal loading stress. This trend indicates a reassuring visibility of subcritical cracks at design stresses, e.g. eight inches (0.2 m).

2. Aluminum Alloy Spreader Beam

There is unfortunately, no extensive data bank for aluminum weldment fatigue with which the new results can be compared. Alternatively the S-N data of this program is compared to that for A-36 steel weldments by its addition to Fig. 4. The comparison is on the basis of outer fiber stress since the base material of the 6061-T651 I-beam has a yield strength (38 ksi) close to that for A-36. The design allowable of the weldment is lower, based on a solution anneal around the weld; about 20 ksi yield strength is typical here. However, the actual limit load of the aluminum beam showed the softening effect to be localized. It is immediately apparent that the fatigue endurance, on this basis, is much less satisfactory. This is not surprising since it is well known from Pearson's work that the rate of fatigue crack propagation at given value of crack load excursion (ΔK) varies inversely as the elastic modulus E. Generally, fatigue crack growth rate varies as about the third power of $\Delta K/E$. For given stress loading level on the beam, hence given ΔK, the growth rate in aluminum should be some 3^3 or about 30 times as fast as in steel. This is indeed, the magnitude and direction of separation of the fatigue endurance values in this data (Fig. 4). As with the steel beam, a simulation of proof test load excursions could not be discerned, nor a deleterious effect of the intermittent salt spray environment. However, the diminished endurance puts the design load/use envelope much closer to the endurance limit boundary.

3. Wire Rope Pendants

As a basis of reference, a number of complete specimens were cycled at constant load excursion to establish ordinary fatigue endurance. The cyclic life data, Fig. 6b is comparable to results of Reemsnyder [24] on various IPS wire rope of similar construction. Recall that in this test, replicate pendants are simultaneously loaded. When the specimen fails the effect on one of the pendants was devastating, Fig. 7. Strangely the other, although subjected to cyclic loading which should have brought it to the verge of failure, showed no external signs of damage. To assess internal damage, the eye splices on several fatigued pendants were cut off, the free length then unwound by strands and thence wire by wire. It was found that the wire rope core

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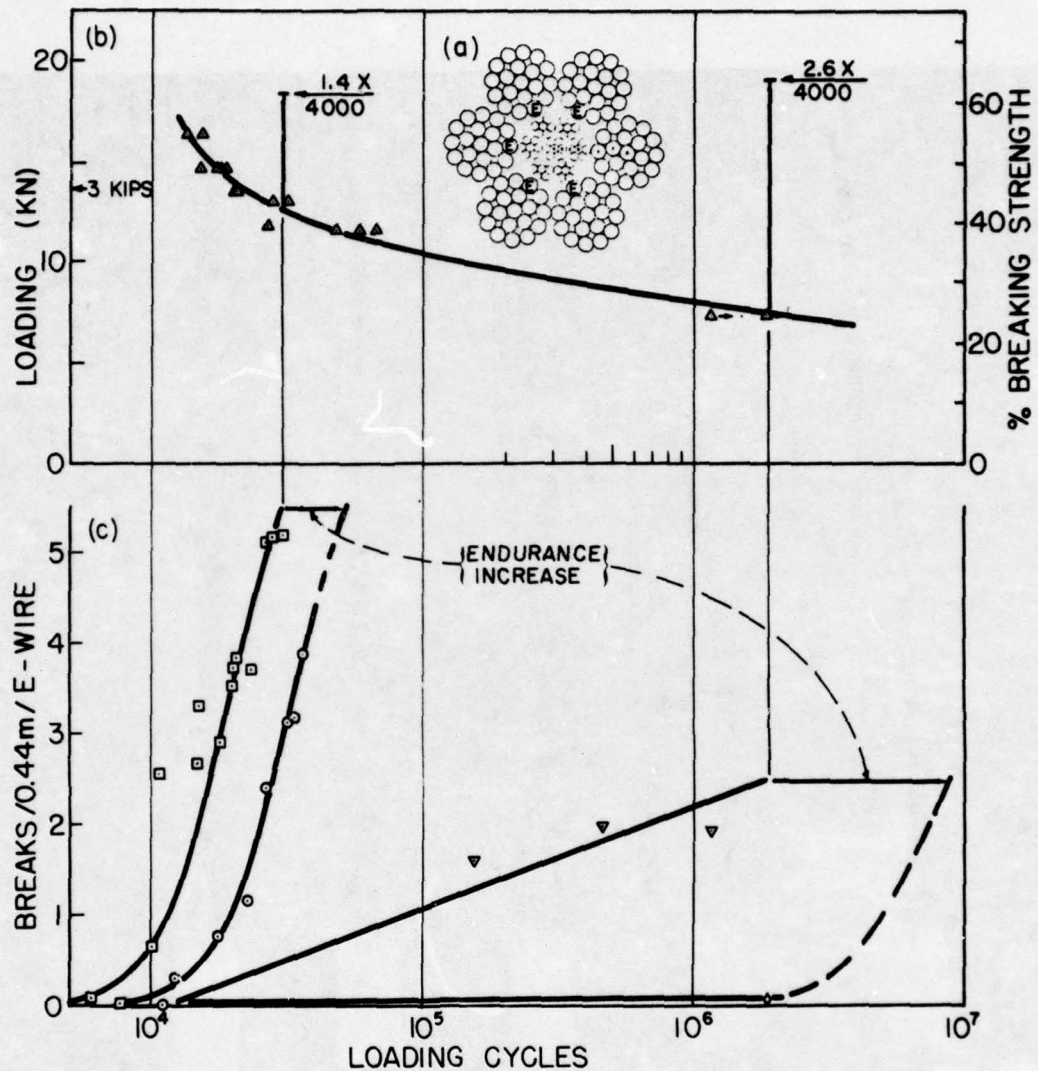Fig. 6 — Cross-sectional view of 6 × 19 IPS wire rope (a) which failed after cyclic loading, as shown in (b). The accumulation of internal wire fissures (c) is found to be retarded by imposing periodic overloadings, to degree shown in (b). Symbols in (c): ⊡ is break count for steady fatigue at 43% breaking strength; ⊙ that with 1.4X overloads applied each 4000 cycles; ▽ for fatigue at 24%; ◇ with 2.6X overloads.
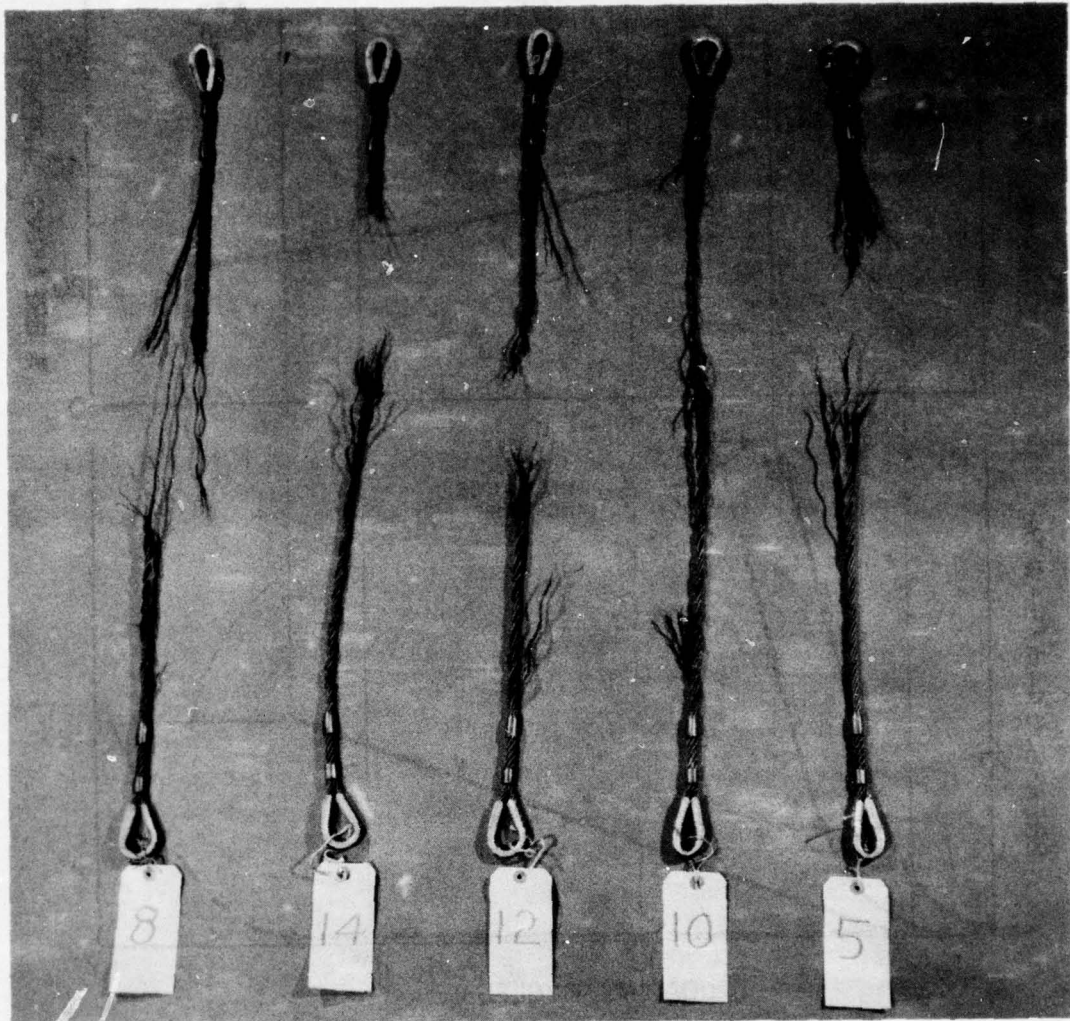
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Fig. 7 — Appearance of wire rope pendants
which failed during endurance tests

and the other wires of the strands, E-wires of Fig. 6a, were highly fragmented. However, the internal wires of the strands, generally survived intact. Examination of the fissured ends of the E-wires in a scanning electron microscope by W. H. Cullen, Fig. 8, showed the breaks to have the appearance of low stress, high cycle fatigue cracks, except for a small residual ligament which failed by ductile rupture. The cracks were associated with fretted zones where the E-wires contacted the core strand. Generally however, the cracks did not initiate in the fret zone but diametrically opposite it, presumably the result of greater tensile stress there from bending of the E-wires over the hard central core.

An attempt was made to quantify the degree of internal fatigue damage. Each piece of broken wire, that is each E-wire and core wire, was painstakingly removed and laid out in order on a sheet of paper, Fig. 9. The length of each fragment, hence the position of the cut off 17.2 in. (372 mm) length, was measured and recorded. Generally, the mates to broken pendants were highly fragmented, with E wires and core wires sustaining some 5 to 10 breaks each in the specimen length; fragments 1.0 in. (25.4 mm) to 2.0 in. (50.8 mm). This is still greater than the pitch distance of E-wire contact against the core 0.6 in. (15.2 mm) which should represent a minimum for E-wire fragment length in view of the contact fretting source of wire fracture.

The extreme disintegration of the "exhausted" pendants led to the question of how this condition developed. Accordingly, a series of pendant pairs were cycled to selected fractions of the expected total endurance. Of each pair, one was dissected in the manner described above; the other was loaded to failure to determine its remaining strength. The break count of the E-wires at two cyclic load levels, 12.8 kN (43% of break strength) and 7.24 kN (24%) is shown in Fig. 6c. At the higher load, the breakup begins at some 25% of the total life and increases steadily to the failure density. At the lower load a relatively earlier onset of wire breakup is indicated, although here, because of the great endurance, only a few data points could be collected.[*]
As break density increased they became randomly distributed along specimen length, suggesting little influence of the termination splice. The breaking strength was found to correlate with wire break density, decreasing about 4% per average E-wire break per meter of rope length.

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[*] This phase of the work was directed by J. A. Kies, (deceased).
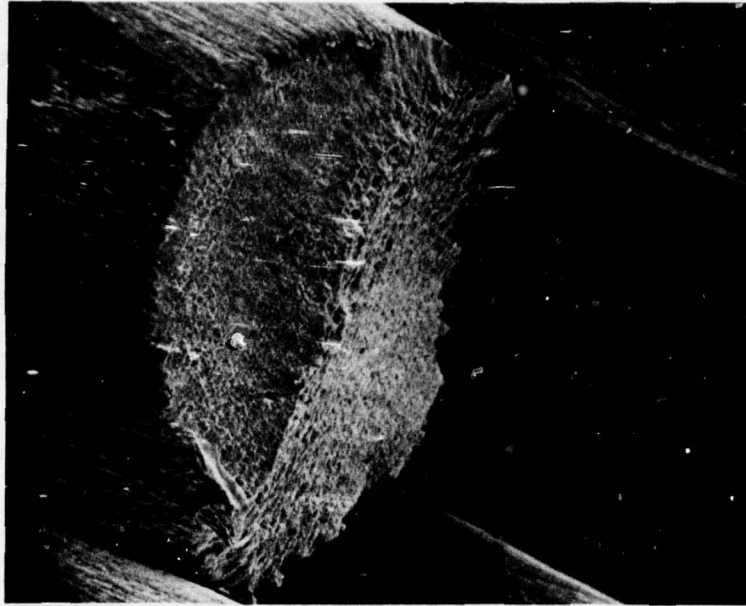
(a) In-situ view of another typical break of a type E wire
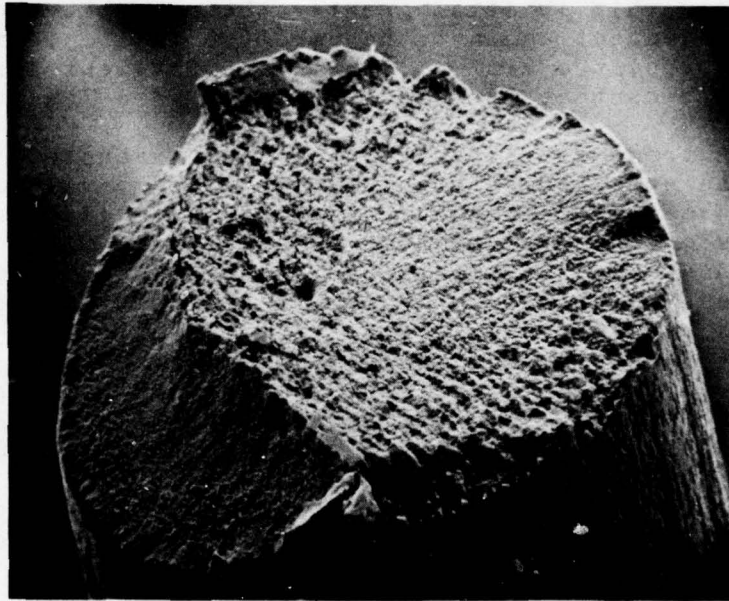
(b) End-on view of the break of Fig. 4(a)

Fig. 8 — An SEM photomicrograph of a typical wire break shows an appearance characteristic of fatigue crack propagation; the flat portion, with terminal rupture along a shear plane, the slant portion. Cracks tend to initiate away from the fretting zone of contact between the wire rope core and outer strands.

Fig. 9 — Disassembly of one of the six outer strands of a 6 × 19 pendant, cycled to near its endurance limit (17,330 cycles at 3650 lb) showed some 50 breaks in the 12 outer wires (of 19) of the strand.
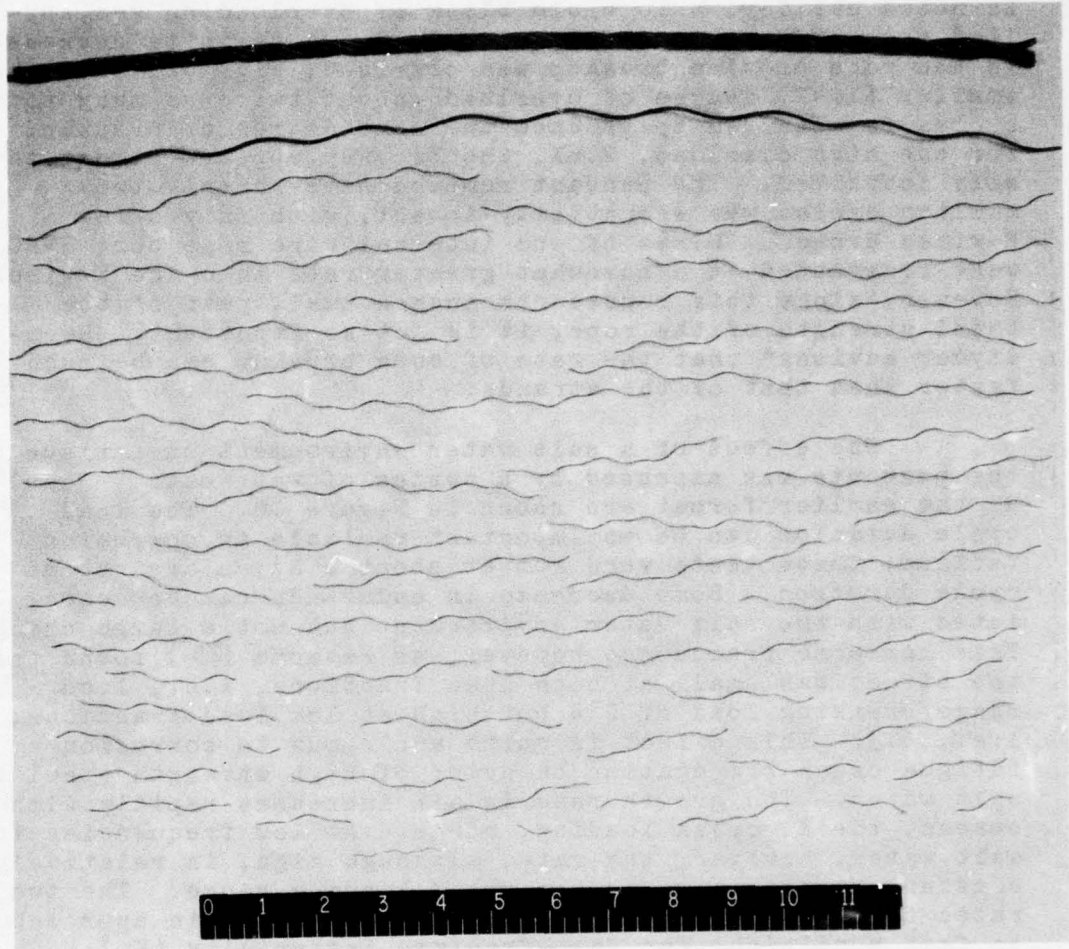
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Effects of proof test loading on the wire breakup were then investigated. A factor of 1.4X overload was applied to the pendants under 43% fatigue load, 2.6X to those under 24%, bringing both to about 62% of breaking strength. As noted earlier, a 10 cycle block of overloading was applied each 4000 cycles of regular duty. A definite decrease in the rate of wire breakup was observed, Fig. 6c. For the smaller (1.4X) degree of overload, about twice as many total cycles is required to produce the same degree of breakup. For the high overload, 2.6X, the breakup appears almost totally inhibited. The pendant removed after nearly two million cycles was essentially intact, with only a few E-wires broken. Wires of the internal wire rope core IWRC were fragmented at a somewhat greater rate than the E-wires. However, since this represents such a small part of the total strength of the rope, it is not so important. Reemsnyder advises* that the rate of core breakup can be much faster than that of the strands.

The effect of a salt water environment on fatigue of the pendants was assessed by a series of wet tests. Results in the earlier format are shown in Figure 10. The load cycle duration can be an important variable in corrosion fatigue; these tests were run at about 1 Hz: a one second cycle duration. Some decrease in endurance can be associated with the salt water environment but not a large change. This has some precedence however, as Matanzo [25] found that the effect was small at high load fractions, i.e., load range/breaking load of 0.4 but high at low load fractions, i.e., 0.3. This effect is quite analogous to corrosion fatigue crack propagation behavior of high strength steel in salt water. The growth rate in air increases rapidly with stress, the K, crack loading, range. At low frequencies in salt water, however, the rate, although high, is relatively constant, insensitive to the crack loading range. The two rates tend to converge at high loading levels, in approach to fast conditions for fast-fracture instability (K_c). Matanzo's cyclic frequency was high, 70 Hz, yet the environmental effect is quite large. What it would be with a longer cycle time is an important question for the NSP application, which requires further investigation.

As with the air environment, overload benefaction of fatigue life was sought in the sea-water tests. There does appear to be some increase in endurances, and corresponding delay in the interval breakup of the rope. However, the data sampling is too limited to quantify the effect, Fig. 9c.
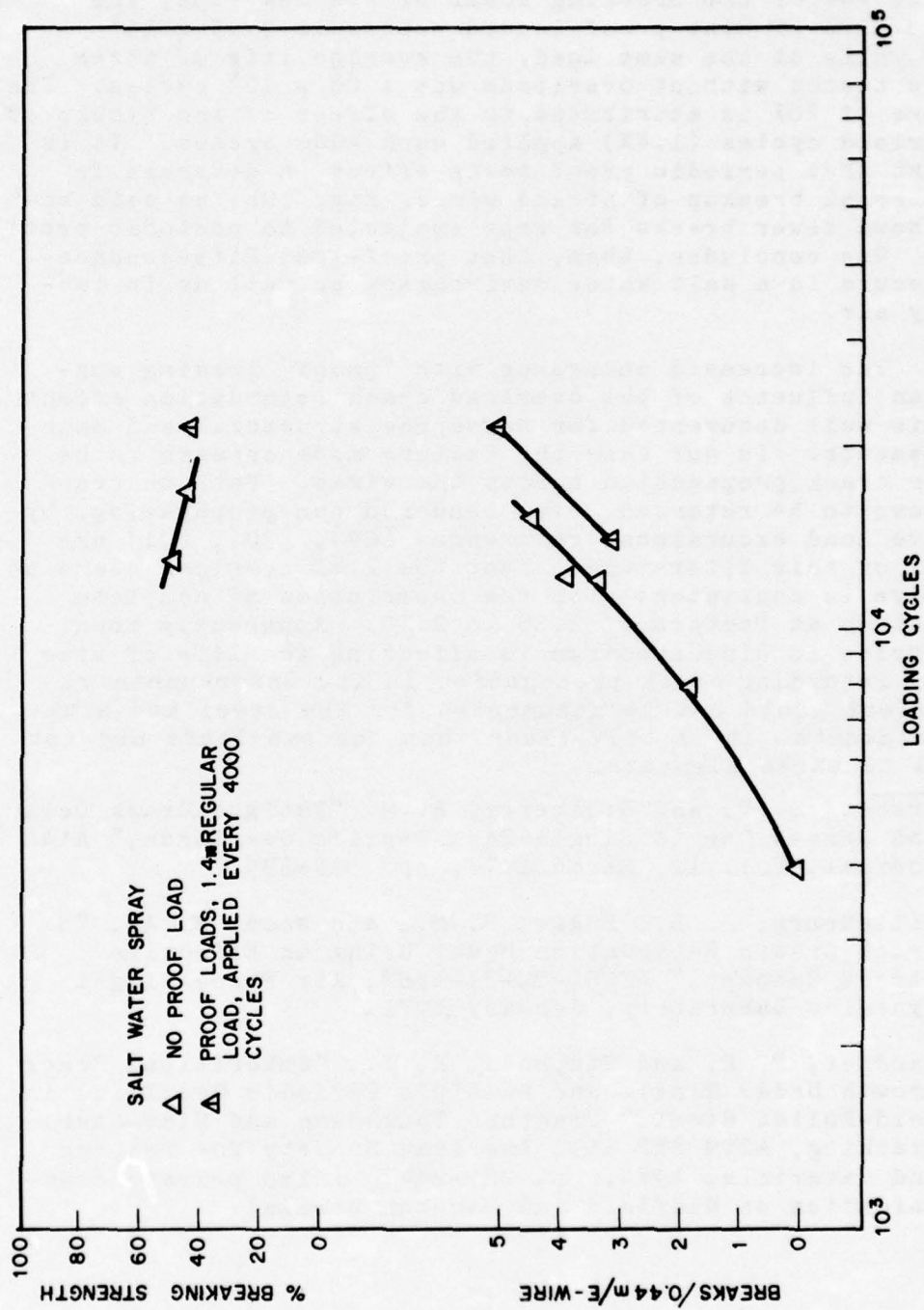
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* H. S. Reemsnyder, private communication.

Fig. 10 — Fatigue strength (a) and wire break density (b) plotted as a function of loading cycles of 6 × 19 IPS wire rope tested in a salt water spray environment

The limited amount of endurance data for wire rope tested in salt water is plotted in Figure 10a. At cyclic loads to 44% of the breaking loads of the new rope, the average life of nine proof-loaded cables is 2.15×10^4 cycles while at the same load, the average life of three samples tested without overloads was 1.68×10^4 cycles. The increase of 28% is attributed to the effect of the blocks of 10 overload cycles (1.4X) applied each 4000 cycles. It is apparent that periodic proof tests effect a decrease in the internal breakup of strand wires. Fig. 10b, as said before, shows fewer breaks for rope subjected to periodic proof loads. One concludes, then, that proof-load life-enhancement occurs in a salt water environment as well as in laboratory air.

The increased endurance with "proof" loading suggests an influence of the overload crack retardation effect which is well documented for aerospace structural and machine elements. In our case the failure mode appears to be fatigue crack propagation across the wires. Fatigue cracks are known to be retarded, even rendered non-propagating, by positive load excursions; references [29], [30], [31] are typical of this literature. That the 2.6X overload seems so effective is consistent with the observation of complete retardation at factors of 2.35 to 2.75. Apparently then, the service loading spectrum is affecting the life of wire rope by retarding crack propagation in it. Unfortunately, this effect could not be documented for the steel and aluminum weldments. It is only clear that the overloads are not harmful to these elements.

29. Probst, E. P. and Hillberry, B. M. "Fatigue Crack Delay and Arrest Due to Single Peak Tensile Overloads," AIAA Journal, Vol. 12, March 1973, pp. 330-335.

30. Willenborg, J. D., Engle, R. M., and Wood, H. A., "A Crack Growth Retardation Model Using an Effective Stress Concept," AFFDL-TM-71-1FBR, Air Force Flight Dynamics Laboratory, January 1971.

31. Gardner, F. H. and Stephens, R. I., "Subcritical Crack Growth Under Single and Multiple Periodic Overloads in Cold-Rolled Steel," Fracture Toughness and Slow-Stable Cracking, ASTM STP 559, American Society for Testing and Materials, 1974, pp. 225-244. (Also private communication on Hadfield and Mar-ten steels).

4. Wire Rope Inspection

During this program an effort was made to find a nondestructive method for detecting the internal breakup of the wire rope. Standard X-ray NDT proved ineffective. Using the wire rope as a core element in a high frequency bridge circuit showed some effect, but it was difficult to reproduce. At a progress review for the NSP 2720, one member of the reviewing staff, Mr. Joseph Hersch, suggested that simply back-twisting the rope might separate the strands sufficiently to allow the fragmented core to be visually observed. A trial at NRL showed this to be a viable technique. Not only are the core and broken E-wires made visible, but they tend to be dislodged to an extent which makes them difficult to ignore. A manual tool for performing the "back-twist" inspection has been designed and a working model built, Fig. 11. It provides two self-locking pliers, padded to clamp onto the 1/4 in (6.4 mm) rope. While one end remains fixed, the other is constrained to permit rotation of 135° in 10 rope diameters between the clamps, which is sufficient to expand the strands yet not permanently deform the rope. Preliminary data indicated little effect of this test on the fatigue life provided that the untwisting does not reveal, hence dislodge, broken wire ends. Normally one should remove rope from service once broken strand wires are observed so that this is not a damaging procedure, although early breakup of core wires might not warrant replacement. Other results show the back twisting technique to be almost as sensitive as disassembly of the rope for early detection of E-wire and core breakup.

VIII. DISCUSSION

It remains here to reexamine the initial questions in view of the foregoing results. What is the safest "factor of safety" for design stress allowables for missile handling equipment? What is the safest structural verification and re-verification policy, particularly as pertains to proof test loading? In foreboding such extreme consequences of failure there is a natural tendency to seek security in the greatest safety factors, the most rigorous verification procedures. But none of these things is without trade-offs in effectiveness. Structurally inefficient design, too little of available material strength utilized, can be unsafe because of excessive weight and size, and consequent unwieldiness. Moreover, one cannot legislate good design by factors of safety, necessary as these are. What constitutes "good" design defies simple definition. It involves sound engineering practice. judgement experience, as well as intangible qualities that border on artistry: a feeling for design; and ingenuity in coping with complex, interrelated
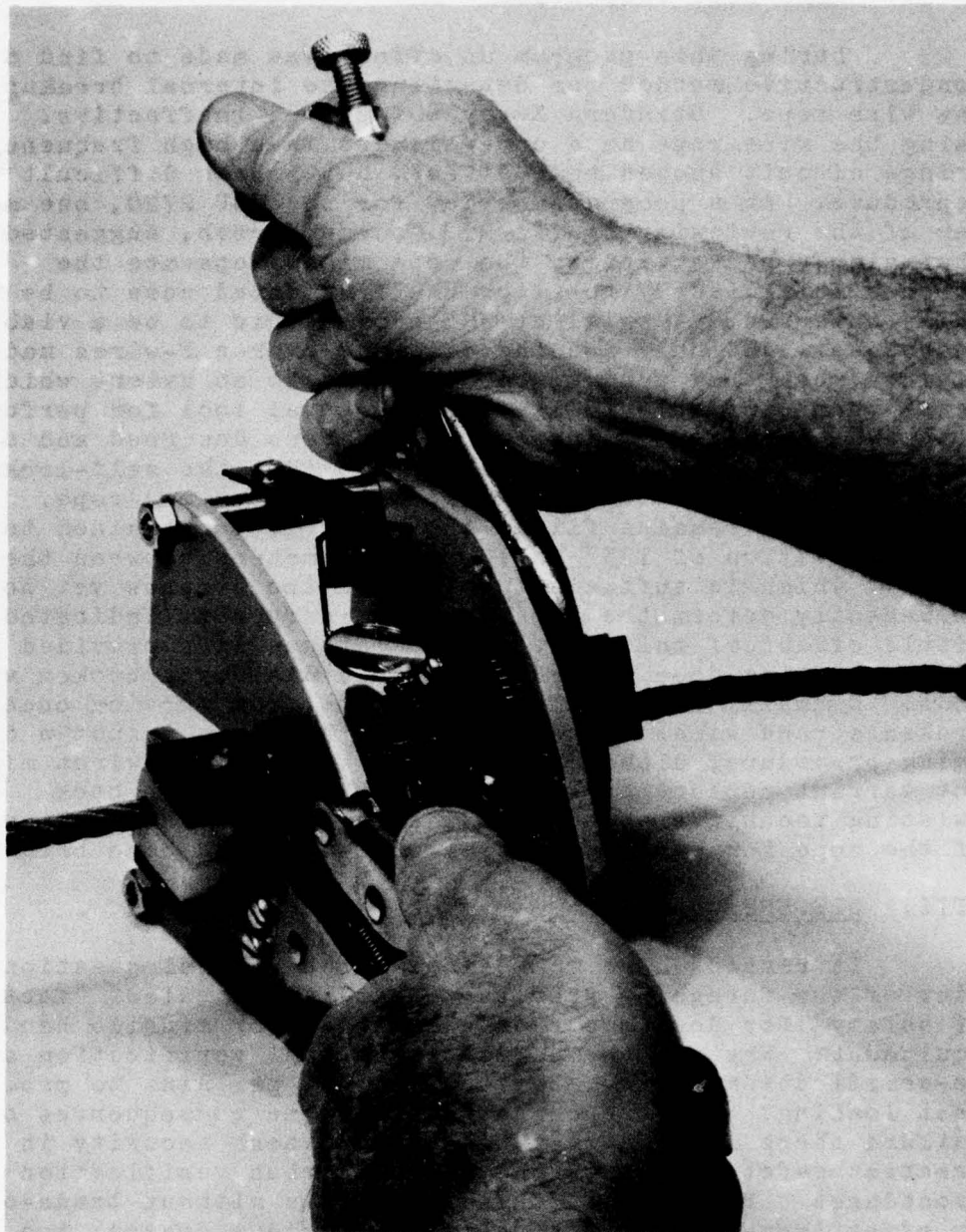
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Fig. 11 — A manual device for back-twisting wire rope to expose internal fissures is shown in operation. Dislodged wires in a section previously tested appear to the left of the device on the pendant.

yet often contradictory requirements. It is useless to
believe that "good" design can be "mandated" by specifica-
tion. Nonetheless, design allowables are necessary safe-
guards, and a part of good design.

An important consideration in setting design allowables
is the degree of perfection of the structure, which varies
greatly with the required function of an item. But given
a class of equipment, the success with which the design/
fabrication has been executed should influence its load cap-
ability. Assurance of proper fabrication is the goal of NDT
inspection procedures, of which the proof loading is an
essential aspect. Better NDT should gain increased preci-
sion in setting design allowables. Hence the two aspects
should be considered as complementary. As it turns out,
the degree of proof test overload required to retard crack
propagation is a factor in setting design allowables, since
the structure must support the proof loading without damage.

1. Benefits of Period Proof Test

Recall here from Sec. IV that present NSSPO-SSE
recertification policy requires proof loading every six
months on shore based equipment and every eighteen months,
or coincident with the shipyard time, for shipboard equip-
ment. A proof load 2.0 times the design load is specified.
Typically, a piece of gear might be used hence loaded, 10^3
times in the maximum duty period. Of what benefit is proof
loading to the safety of this equipment? Let it here be
clearly recognized that whatever the objective effects,
there are important intangible benefits. Everyone feels
more secure in working with gear that has been recently
checked, hence proved capable of handling its load. Of more
tangible benefit is the occasion to check for general signs
of deterioration, damage or wear: rust, corrosion, bare spots
in the paint, dirt, need for lubrication, cleaning, etc.

a. Crack Growth Retardation in Weldments

A more quantifiable benefit derives from the
retardation of fatigue cracks due to proof overload. It is
now known that load peaks interposed in a series of constant
load applications will retard the rate of fatigue crack pro-
pagation. Actually the growth rate in the early cycles after
such overload is somewhat greater than before, but when the
crack progresses further into the plastic zone produced by
such overload, the rate declines. At somewhere between
overload ratios of two to three, complete retardation has
been observed: 2.35 is a well documented factor for high
strength aluminum by Probst and Hillberry [29], 2.0 is used

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in the Willenborg retardation model [30], 2.35 and 2.75 was found by Gardner and Stephens for Hadfield and ManTen steels respectively, but little beneficial effect in cold rolled steel [31]. The results of this test program on the steel and aluminum spreader beam weldments are, as indicated earlier, inconclusive as to benefits of proof loading. It was evident, however,that the proof "overloads" were not harmful. An exception here, it should be cautioned, is one case in which the "proof" load approached the limit load for the beam, causing failure during the proof load cycles.

What is the effect of overload peaks on wire ropes? Reemsnyder [24] has compared the endurance of wire rope pendants given an initial 2X overload with those as received, as a reference. No significant benefit was detected. In NRL's program however, the overload was repeated every 4000 cycles. With 1.4X overload, a marked dimunution in wire break density was found, with enhanced endurance. With 2.6X overload the break up of wire was almost completely suppressed, with a corresponding large increase in fatigue endurance. Apparently, the fatigue crack propagation across individual wires of the strands is indeed retarded by the overloads, presumably for the same reason as for the larger rigid elements. The 2X overload, if not repeated too frequently, does appear beneficial. Apparently 2.4X would be even better for this purpose.

What is the effect of proof loading on the strength of a structure at a subsequent loading at a different temperature? We are concerned here with the low temperature brittleness of the mild steel weldments as in winter or artic service. The power plant pressure vessel community has addressed this question. It is found, papers by Brothers and Yukawa [32], Harrison and Fearnehough [33], Andrews [34],

32. Brothers, A. J. and Yukawa, S., "The Effect of Warm Prestressing on Notch Fracture Strength," Journal of Basic Engineering, Trans. ASME, Series D, Vol. 85, No. 1, March 1963, pp. 97-104.

33. Harrison, T. C. and Fearnehough, G. D., "The Influence of Strains Due to Proof Loading on Welding Embrittlement," Metal Construction and British Welding Journal, 1969, No. 1, pp. 476-478.

34. Andrews, W. R., "Effect of Loading Sequence on Notch Strength of Warm Prestressed Alloy Steel, ASME Paper 70-PVP-10.

Succop, Pense and Stout [35], that after "warm prestressing", that by proof loading at temperatures above the use temperature, of a crack specimen, subsequent fracture failure at a lower temperature requires a load substantially greater than expected for this temperature. Unfortunately, the failure load is not always as high as that of "warm-prestress", particularly if temperatures are reduced to levels well below the "nil ductility" transition temperature (NDT) of a steel. Also Yukawa cautions that the directional pattern of stresses must be the same at both temperatures, which is often difficult to duplicate in reactor vessels subjected to large thermal stress. Also the warm temperature must not be so high that creep induced stress relaxation mutes the protective residual compressive zone at the crack tip. However, neither of these conditions apply to our case. Hence, the proof overload should serve to reduce susceptibility to low-temperature, low-stress, brittle fracture. A caution here is that the warm prestress effect is lost if there is sufficient subcritical crack growth after overload to move the crack out of its protective compressive zone. The crack growth retardation effect of the overload of course, tends to delay the revision to normal fracture toughness.

2. "Non-benefits" of Proof Test Loading

The promise held forth in the fracture mechanics analysis of prooftesting of Tiffany [36] is a guarantee of finite "safe life". No failure in the proof test means cracks no larger than those critical at the proof load can be present. At the use load, the largest of these has a predictable distance to grow before it can become "critical". Unfortunately, later work of Vroman* showed this to be inapplicable to low strength, tough materials. We attribute this to the cyclic strain hardening of the softer materials. As a result, the (crack) loading limit ΔK for an unstably high fatigue crack propagation rate occurs far below K_{Ic}, the fracture toughness in monotonic loading. Typically in mild steel or weldable 5000 series aluminum, the mismatch is by about a factor of three. This means that below overload ratios of three, no additional "safe life" (except for the crack growth retardation effect) can be guaranteed.

*G. A. Vroman, private communication.

35.   Succop, L.N., Pense, A.W. and Stout, R.D.,"Effects of Warm Overstressing on Pressure Vessel Steel Properties," Welding Journal, Aug. 1970, pp. 354s-364s.

36.   Tiffany, C.F., "Fracture Control of Metallic Pressure Vessels," NASA SP-8040, May 1970. Available from National Technical Information Service, Springfield,Va.

Such high overload ratios would seem a practical impossibility since they would tend to require building of very heavy, cumbersome, inefficient structures, i.e., with factor of safety greater than 3.0, to support such loads.

A closely related misconception of "proof test" effects is the expectation that it will cause failure in "defective" equipment. In mild steel structures, enormous cracks can survive proof testing at 2X design allowable stresses. In the fatigued wire rope pendants, the mates to pendants which failed were found to possess about 65% of their initial strength yet too strong to fail the proof test at 40% of initial strength. At smaller fractions of their endurance limit, the breaking strength is even higher. We conclude then, that reasonable proof test loadings will not fail badly deteriorated rope nor cracked solid structures hence, not warn of the unsafe condition.

There has been some concern that a too-oft-repeated application of proof loads could accentuate fatigue damage leading to failure. It seems clear that for the modest loads and total possible number of recertification cycles of this equipment, such damage is negligible.

A most conservative damage accumulation rule, such as a linear cumulative damage rule, predicts little damage for forty cycles, two proofs per year for twenty years. Hence, although proof testing does not provide complete assurance of integrity, it is not a harmful procedure.

3. Optimal Frequency of Proof Loading

There may be practical considerations which mandate a frequency of proof testing greater than that indicated for maximum fatigue life of the equipment. Nonetheless, it could be useful to estimate what is the optimal frequency. The question has been investigated experimentally by Gardner and Stephens [31], who examined the effect of number of use cycles between overloads on the total life of crack propagation specimens. In cold rolled steel, the maximum life occurred with about 5,000 cycles between overloads. In Hadfield steel it tended to peak with greater use between overloads, e.g., 20,000 cycles. The optimal number of cycles between proof test tends to increase with severity of the overload - likely associated with increased size of the overload plastic zone. This contention is consistent with the better performance of the Hadfield steel, whose lower (than CRS) yield strength would entail a larger plastic zone upon overload. Presently available data would suggest, then, that a proof test frequency greater than once every 5000 cycles will be of greater than optimal frequency. However,

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a higher frequency, in the context of this equipment usage and recertification policy, should not be harmful because the total of such overload cycles is so small.

4. Statistical Confidence for Fatigue/Strength Data

Estimating the reliability limits associated with design allowables relative to experimentally determined limits of failure requires knowledge of the statistical distribution of data points for a given determination. It is difficult to quantify the probability of fatigue failure, especially with regard to establishing a guaranteed number of cycles before which failure cannot occur. A recently published paper by Mann and Fertig [37] is applied below to the analogous problem in the case of wire rope fatigue. The same technique could be applied to suitable failure data on weldments in order to determine whether a lower bound on weldment life can be established by statistical means. However there is a scarcity of data on weldments; the testing is slow and costly. The effort of this program adds but slightly to the major collection of Gurney [21] whose data corroborates ours rather than the reverse.

Actually, there is surprisingly little scatter in the fatigue life of weldments of a given class, a point made by Gurney. This can be rationalized if one views the weld as a prompt and reproducible crack starter. From the physical point of view, it should be borne in mind that the S-N curve for the class of welds where few cycles are required for crack initiation, is essentially an integration of the growth rate da/dN vs stress intensity factor excursion ΔK relationship. This da/dN (ΔK) relationship is derivable from physical (mechanical) properties of the materials: its cyclic strain hardening rate and the strain rate sensitivity of flow stress. These properties have little variance in value, much like the elastic modulus or density which determine acoustic wave velocity. The growth rate of a fatigue crack of known loading, can vary only within relatively narrow limits. The initial crack size is bounded by NDT procedures and the terminal critical size by fracture toughness. There is a finite path. It is physically impossible for the rate to be infinite. There is accordingly a very low probability of premature failure.

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37. Mann, N. R. and Fertig, K. W., "A Goodness-of-Fit Test for Two Parameter vs Three Parameter Weibull; Confidence Bounds for Threshold," Technometrics, Vol. 17, No. 2, May 1975, pp. 237-245.

The status of design for fatigue of aluminum weld-
ments is much more primitive. Relatively little data has
been published and there is no widely accepted standard of
design allowables for fatigue. If we accept the point of
view, as in steel, that the worst-case life is derivable
from the da/dN (ΔK) relationship, then the fatigue allow-
ables for given life will vary with the elastic modulus E,
and hence will be only one-third of those for steel. This
would narrow the margin of conservatism compared to that
inherent the current static design allowables for steel.
However, even considering this, it is still likely that 10^4
cycle service would not entail danger of fatigue failure as
indicated in Figure 4 which indicates that the corresponding
stresses are well above those which would be used as design
stresses.

As with the aluminum weldments, there is no accepted
design standard for fatigue of wire ropes. However, the
extensive investigations of Reemsnyder [24] can provide a
basis for estimation. It is clear from this work that the
"S-N" curve varies markedly with type of rope and end term-
ination used. In some cases, however, enough data are
available at a given set of conditions to provide useful
results. For example, in his Fig. 1a, (lines 10 and 12-17)
Reemsnyder [24] gives 33 data points for uncoated improved
plow wire rope with IWR core and sleeve grips. The Mann and
Fertig test [37] was applied to these data to distinguish
whether they are from a Weibull distribution with 1) zero
location parameter, or 2) positive location parameter, i.e.,
a guaranteed number of cycles before failure. The hypo-
thesis of zero location parameter was rejected at about the
1% level of significance, that is, there is only a 1% chance
that these data are from a distribution with no guaranteed
number of cycles before failure becomes possible. The Mann
and Fertig paper also provides a technique for obtaining a
lower confidence bound on the guaranteed number of failure-
free cycles and a computer program is being written to per-
form the necessary calculations. Pooling more of Reem-
snyder's data has allowed establishment of a position loca-
tion parameter at a smaller level of significance and a more
optimistic lower confidence bound. It must be emphasized,
however, that all of these results are based on the assum-
ption of a Weibull distribution. This assumption has often
been made for similar applications but, like all other math-
ematical models, it is only an approximation to reality.
Moreover, it is noted that it is impossible to obtain an
absolute "guarantee" for the guaranteed number of failure-
free cycles. In other words, for any finite amount of data
with non-zero dispersion it is impossible to reject the
hypothesis of immediate failure (at or before the first
cycle) with probability one. (The foregoing analysis is

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due to H. E. Ascher).

5. Recommended Design Stresses and Proof Test Frequency

The design practice indicated in the proposed speci-
fications is tantamount to using a minimum yield factor of
safety of 2.3 as discussed in Sec. III and shown in Table II.
The results of this study strongly suggest that a modest
increase in this factor, or decrease in maximum allowable
stress, would be desirable. A basic safety factor as low as
1.15 on yield stress is tolerated, with concommittant qual-
ity and inspection for weight-critical structures for the
aerospace applications. Weight-handling equipment need not
be nor rarely in fact is, designed to such low safety fac-
tors. The generally accepted industry standards for this
class of equipment require higher safety factors, as summa-
rized in Table II. Higher safety factors will also permit
safe proof loading to 2.35 to 2.5 times the rated load, a
practice seen to most beneficially delay the continuance of
crack propagation due to repeated application of the rated
load. It is for these reasons, then, that the higher mini-
mal safety factors shown in Table III are recommended for
NSP surface support equipment. These recommendations are
consistent with industry and DOD standards, Table II,
except for MIL-STD-1365. When MIL-STD-1365 allowable
stresses are not inconsistant with required structural effi-
ciency, design to such levels is encouraged. In any case,
adherance to established industry standards, such as pre-
viously noted AISC handbook [11], Aluminum Association Spec-
ification [26] and the Military Standardization Handbook
MIL-HDBK-5B [7] is appropriate. With allowable stresses
limited in this way, frequency of proof loading greater
than once in 5000 use cycles can hardly be justified from a
fatigue damage viewpoint, as discussed earlier.

On wire rope elements, a rated load limitation of
20% minimum breaking strength, the industry practice [9] is
recommended. Judging from typical fatigue life data, the
degree of safety provided by this factor in air environment,
is about equivalent to 33% of yield strength (or 25% of
ultimate) on mild steel weldments. In this way, the struc-
tural elements will yield, hence warn of overloads, before
catastrophic failure of the wire rope. Regarding aluminum
weldments, static strength allowables commensurate with that
for steel are possible. Here, however fatigue allowables
must be greatly reduced. A reduction in proportion to the
relative values of Young's modulus, about 1/3, is indicated
by crack propagation behaviors.

6. Continuing Work

The Strategic Systems Project Office, NSPO, is
supporting additional work at the Naval Research Laboratory
(NRL Code 8432) toward optimization of wire rope material
and construction for ordnance weight-handling applications.
The endurance lives of four different types of 1/4 in
(6.4 mm) rope are being determined. Samples of these ropes
will be subjected to portions of their endurance lives and
then evaluated for damage. This will consist of counting
the number of breaks in the individual wires comprising the
rope and in determining the residual tensile strength of the
corresponding partially fatigued pendants. This investi-
gation will contribute toward the optimization of rope
material, construction as well as operating pattern, for
ordnance weight-handling applications.

NSPO is also funding work at NRL to evaluate
currently available RF high voltage insulator links. These
are used to electrically insulate the lifting machine from
the load being handled, thus providing electrical shock
protection for both personnel and load.

IX. CONCLUSIONS

Results of this study and experimental program have
led to the following conclusions relevant to the design
and certification of TRIDENT C-4 surface support equipment.

1. A 2X proof test load will not assure a measure of
"safe life" in fatigue in low strength steels and alum-
inums. This effect is associated with their high fracture-
toughness relative to fatigue-toughness limit.

2. Repeated loading of wire rope may result in severe
fragmentation of internal wires which is not apparent in
visual inspection of the rope exterior. Ordinary NDT
inspection techniques and the 2X proof load may not detect
this condition.

3. Proof test loading tends to retard fatigue crack
propagation in wire rope.

4. The optimum effect of proof loading occurs when
a proof test is applied at a load level 2.35 to 2.5 times
the use load and is applied at frequencies no greater than
every 5 thousand cycles of service loading.

5. To preclude post proof test cold-cracking, proof
loads should be applied no sooner than 48 hours after
welding.

6. A newly devised back-twisting method and tool for examination provides a useful NDT method for small diameter flexible wire rope.

7. The proposed new weapons specifications Spec. 63, 64 and 69 are well founded documents. However, some changes in safety factors for design allowables in NSP surface support equipment would appear advisable in view of results of this study.

8. The static design allowables of the proposed specifications are below levels which could lead to fatigue crack failure of the mild steel weldments and IPS wire rope; the fatigue life at design stresses, far exceeds any anticipated service requirements. The case for aluminum weldments is rather marginal, however.

9. Moderate exposure to salt air environment is not expected to materially degrade steel and aluminum alloy structures. However, wire rope elements should be watched for signs of excessive corrosion.

X. RECOMMENDATIONS

1. Weight-handling equipment should be subjected to a proof load between 2.35 and 2.5 times the use-load as part of the acceptance procedure of new, repaired, refurbished, or significantly altered equipment. The proof load test *should only be repeated every* 5,000 to 20,000 loading cycles thereafter. It is to be noted that extreme service for NSP equipment involves a use frequency of one cycle per working day, or 5200 cycles in 20 years, hardly a cause for fatigue damage.

2. Equipment subjected to proof loads should be examined afterwards for possible damage, using visual and other appropriate non-destructive inspection techniques. This examination should be part of the proof test procedure.

3. Weight-handling equipment should be designed in accordance with the criteria proposed in Section VIII.5. of this report.

TABLE III. DESIGN RECOMMENDATIONS WEIGHT-HANDLING AND SERVICING EQUIPMENT

EQUIPMENT	DESIGN LOAD	MIN. YIELD		MIN. ULTIMATE	
		F.S.	LOAD	F.S.	LOAD
ALL EQUIPMENT OTHER THAN HOISTING EQUIPMENT AND UNFIRED PRESSURE VESSELS	L	2.0	2.0 X L	3.0	3 X L
HOISTING EQUIPMENT EXCEPT CABLE ASSEMBLIES	2W	1.5	3.0 X W	2.0	4 X W
HOISTING EQUIPMENT CABLE ASSEMBLIES	W	N/A	N/A	5.0	5 X W
UNFIRED PRESSURE VESSELS	P	N/A	N/A	4.0	4 X P

L = Maximum load expected during normal life, including dynamic load components.

W = The gross weight which must be lifted, i.e., rated load, safe working load, capacity.

P = Maximum working pressure for unfired pressure vessels.

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